Heating · Piping and Air Conditioning

Vol. IV February, 1932 No. 2

Calculation of Friction Loss *in a* Multiple Loop Header System

By Max W. Benjamin*

In choosing one of several possible lay-outs of a steam piping system, the determination of pressure loss plays an important role. The lay-out may be perfectly satisfactory so far as stress considerations and operating reliability are concerned, but if pressure losses are large, changes in pipe sizes or lay-out will be required.

In this article the method of computing pressure losses in multiple loop header systems is shown and demonstrated by the lay-out at Delray No. 3. Particular attention is given to loss through fittings.

AMONG the many perplexing problems involved in the design of piping systems the question of friction loss is one of considerable importance.

Factors Which Influence the Lay-out

It happens frequently that several possible lay-outs will be made and the choice of the design depends upon many factors such as ability to operate satisfactorily under various emergency conditions, flexibility for the accommodation of thermal expansion without undue stress, velocity of flow and friction loss, and first cost. Consideration of stress and operating reliability are of primary importance, but it sometimes occurs that a system which is satisfactory in those respects will result in pressure losses so large that changes in pipe size or lay-out or both will be needed. These changes in turn influence the first cost and expansion characteristics.

An illustration of these conditions is given by the design of the main steam piping in Delray power house No. 3 of The Detroit Edison Company. In this particular case the allowable pressure drop over the system is limited by the maximum working pressure of the boilers. That is, at the boiler end, any increase in pressure demands an increase in drum thickness and expensive design changes, while at the turbine end a decrease in pressure below 375 lb. gage the throttle results in increased turbine heat consumption. It thus became necessary to check each lay-out for pressure drop. The system finally chosen is represented by the isometric diagram of Fig. 1 and consists of 16-in. 400-lb. *A.S.A.* pipe and 600-lb. *A.S.A.* fittings throughout, with the exception of the boiler leads, which are 12-in. 400-lb. *A.S.A.* pipe with 600-lb. *A.S.A.* fittings.

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Pressure Loss Through Fittings

Attention is directed to the large number of valves and fittings used in this lay-out, for here is one of the most important items of the whole problem, since large friction losses occur in them. Some interesting data on the flow of steam through 14 in. x 16 in. x 16 in. tees at

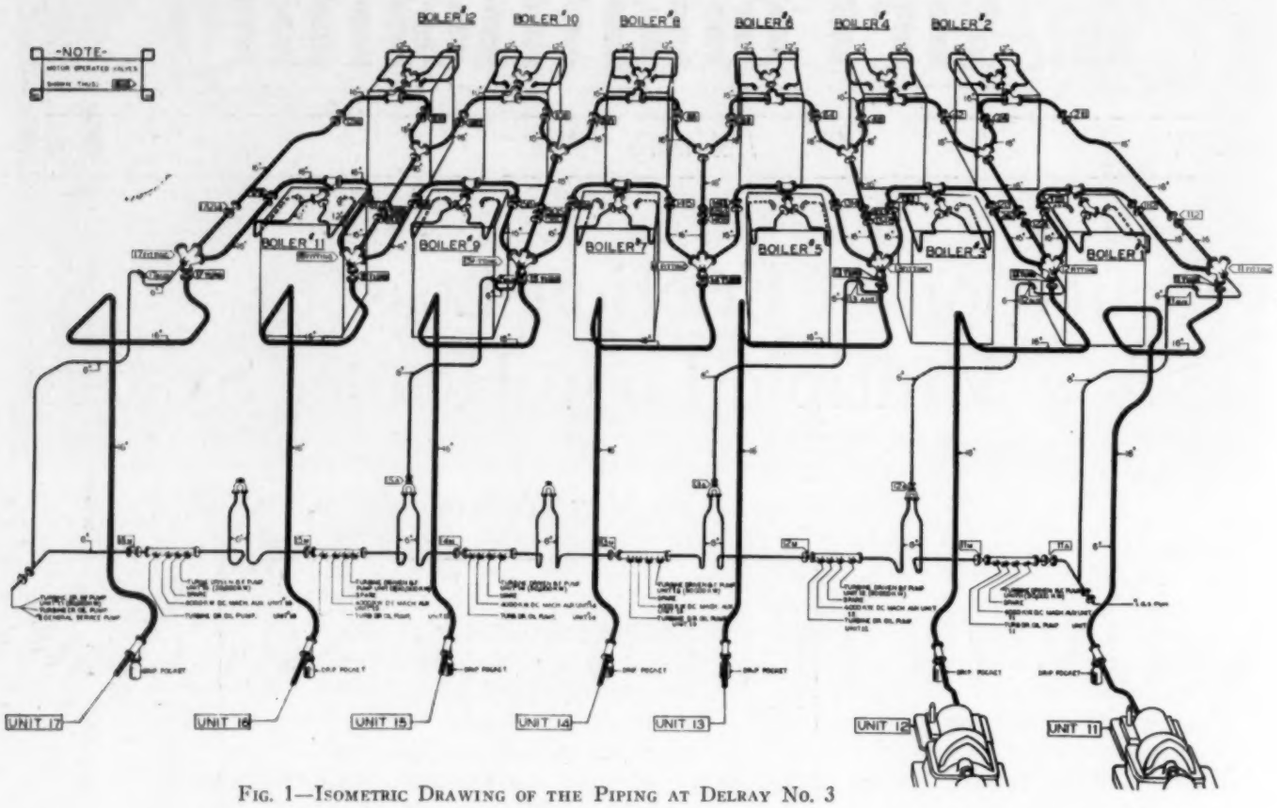


FIG. 1—ISOMETRIC DRAWING OF THE PIPING AT DELRAY NO. 3

Delray P. H. 3 are available. These data have been developed along the lines of tests at the University of München and are given in the appendix. They are not helpful in this instance, however, because similar data on other branched fittings cannot be found, and since nearly all of the fittings used in this particular system are special castings. It was necessary, therefore, to make some assumptions concerning the pressure loss through these fittings. These assumptions are as to the lengths of straight pipe which would result in the same friction losses as are caused by the fittings.

In the case of such fittings as tees and Y's, it has been found convenient, although not strictly accurate, to assign some equivalent length from the intersection of the branches to each flange, and to use such lengths as constants irrespective of the amount or direction of flow through the individual branches. The inaccuracy of this assumption results from neglect of such factors as aspirator action in the case of mixed flow, and "bull head" action in the case of divided flow. Foster's formula[1] for equivalent length was used for standard fittings, and for the special fittings a value of 75 ft. from the intersection to each flange was assigned, which seemed reasonable in the light of tests of somewhat similar fittings at the Trenton Channel plant. Views of the special five-way fittings are shown in Fig. 2. Values of equivalent resistance according to Foster's formula are given in Table 1.

Calculation of Pressure Loss

After deciding on the assumption of the equivalent length of the fittings, the total equivalent length between intersections of the header is calculated and indicated on a schematic diagram. For the Delray system the diagram is as shown in Fig. 3. With this diagram the calculation of pressure loss can be carried out for any desired condition of operation. A method of solving such a problem can best be given by an example.

For a sample problem assume the following conditions:

1. No. *11* main unit and auxiliaries not operating.
2. Boilers *9* and *11* not operating.
3. Rated load on main turbines *12*, *13*, *14*, *15*, *16* and *17*. Steam for each main unit and auxiliaries = 582,620 lb. per hr.
4. All boilers except *9* and *11* steaming at 349,570 lb. per hr. each.

It is desired to know what pressure drop will exist in the header system. (The calculation of pressure drop in the boiler leads and turbine leads is a simple friction loss problem.)

The calculation of the loss over the header system is a trial-and-error process, based on one fundamental fact—that a flow of steam is accompanied by a drop in pressure, the direction of flow being toward the lower pressure region.

The use of **Fritzsche's formula** for the flow of superheated steam at high velocities is recommended for use in this type of problem. For different steam condi-

[1] *A. S. M. E. Transactions*, Vol. 42, p. 648, 1920.

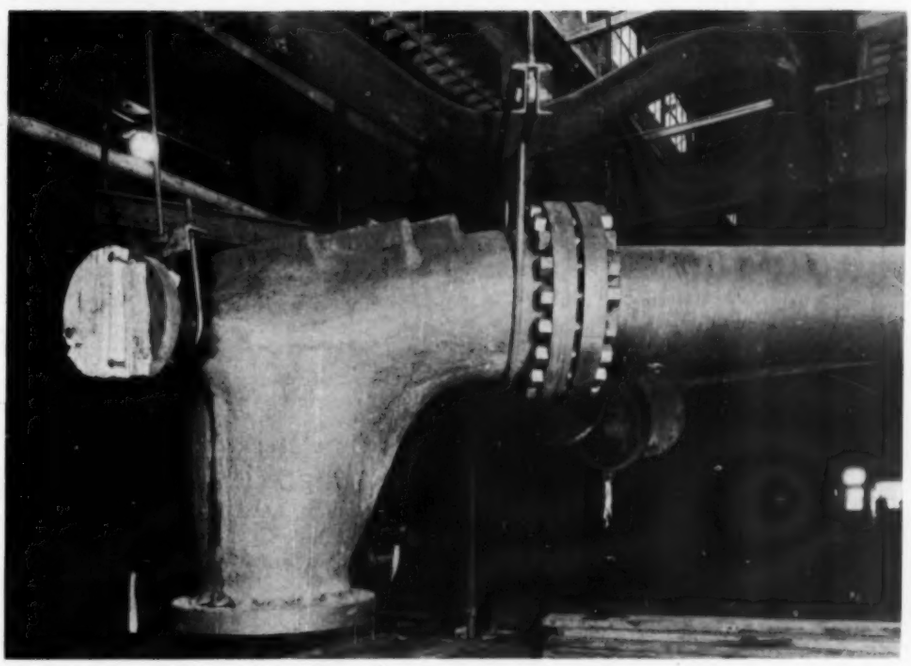


FIG. 2—TWO VIEWS OF NO. 13 FITTING IN THE MAIN STEAM LINE. ABOVE, FACING WEST. BELOW, FACING EAST.

tions or for different fluids (such as water, oil, or air) other formulas should be used. Fritzsche's formula equates pressure loss and flow as follows:

$$p = \frac{0.8\ LF^{1.85}}{yd^{4.91}}$$

where

p = pressure loss in lb. per sq. in.
L = equivalent length of line in ft.
F = weight of steam flowing in lb. per sec.
y = average density of steam in lb. per cu. ft.
d = internal diameter of pipe in inches

Since any very extensive system requires many solutions of the formula it will be found most convenient to **plot a curve** of the above relation, using a scale sufficiently large to allow of pressure drop readings to one-hundredth of a pound per square inch. Such a curve is shown in Fig. 4, though, of course, this reproduction cannot be read with the required accuracy.[2]

In examining the lay-out for a starting point, it is reasonable to assume that boiler pressures in the back row will be higher than those in the front row because the steam will flow from the back row toward the front row and, because the back row is delivering six-tenths of the total steam. It also is noted that in general the steam will flow from the left to the right in Fig. 3. In all probability then, the highest pressure in the header will be at point O. By the same arguments it would appear that the lowest pressure in the header will be at M. Now if the steam flows from O to P and O to N are properly selected the pressure calculated from point M will be the same when going around the back header as when going around the front header. This must be true, for otherwise at some point in the system the steam would begin to recirculate. This point is made obvious in the following calculation of the first trial.

First Trial: For the first trial, assume the pressure at O = 405.80 lb. per sq. in. and steam flow O to B = 45,000 lb. per hr. The pressure drop from O to

[2] To be strictly correct a family of curves should be constructed, covering the densities corresponding to various pressure in the pressure drop range. Actually, however, this is an unwarranted refinement, especially if the total pressure drop over the system is small relative to the working pressure.

TABLE 1—EQUIVALENT RESISTANCE OF FITTINGS, FOSTER'S FORMULA

A. S. A. Rating	Size	Actual I. D.	I. D.$^{1.2}$	Gate Valve	Long Ell or Run of Std. Tee	Med. Rad. Ell or on Run of Tee Rad. 25%	Std. Ell or, on Run of Tee Rad. 50%	Angle Valve	Close Return Bend	Tee Through Side Out	Globe Valve
	Resistance Factor:			0.25	0.33	0.42	0.67	0.90	1.00	1.33	2.00
250	12	12.00	1.00	10.9	14.4	18.4	28.8	39.4	43.7	57.6	87.4
	14	13.19	1.12	12.3	16.2	20.6	32.4	44.1	49.0	64.8	98.0
	16	15.06	1.312	14.4	19.0	24.1	38.0	51.7	57.4	76.0	114.8
	18	17.00	1.519	16.6	21.9	27.9	43.8	59.8	66.4	87.6	132.8
	20	18.88	1.722	18.8	24.9	31.6	49.8	67.8	75.3	99.6	150.6
	24	22.75	2.154	23.6	31.1	39.5	62.2	84.8	94.1	124.4	188.2
400	12	11.75	0.973	10.7	14.1	17.9	28.2	38.4	42.6	56.4	85.2
	14	13.00	1.103	12.1	15.9	20.3	31.8	43.5	48.3	63.6	96.6
	16	14.94	1.300	14.2	18.8	23.9	37.6	51.2	56.9	75.2	113.8
	18	16.81	1.498	16.4	21.6	27.5	43.2	59.0	65.5	86.4	131.0
	20	18.75	1.706	18.6	24.6	31.4	49.2	67.2	74.6	98.4	149.2
	24	22.50	2.125	23.3	30.7	39.1	61.4	83.7	93.0	122.8	186.0
600	12	11.50	0.950	10.4	13.7	17.4	27.4	37.4	41.5	54.8	83.0
	14	12.69	1.069	11.7	15.4	19.6	30.8	42.1	46.7	61.6	93.4
	16	14.50	1.254	13.7	18.1	23.1	36.2	49.4	54.9	72.4	109.8
	18	16.38	1.450	15.9	21.0	26.6	42.0	57.1	63.4	84.0	126.8
	20	18.19	1.643	18.0	23.7	30.2	47.4	64.7	71.9	94.8	143.8
	24	21.94	2.061	22.6	29.8	37.9	59.6	81.3	90.2	119.2	180.4
900	12	11.00	0.900	9.9	13.0	16.6	26.0	35.5	39.4	52.0	78.8
	14	12.13	1.012	11.1	14.6	18.6	29.2	39.9	44.3	58.4	88.6
	16	13.94	1.196	13.1	17.3	22.0	34.6	47.1	52.3	69.2	104.6
	18	15.69	1.378	15.1	19.9	25.3	39.8	54.3	60.3	79.6	120.6
	20	17.44	1.565	17.1	22.6	28.8	45.2	61.6	68.5	90.4	137.0
	24	21.00	1.956	21.4	28.2	35.9	56.4	77.0	85.5	112.8	171.0

$L_e = 43.7\ rd^{1.2}$
L_e = equivalent length of straight pipe, ft.
r = resistance factor
d = internal diameter of fitting, ft.

$B = 5.09 \times 0.02 = 0.10$ lb. from the curve of Fig. 4 and the number of hundred feet equivalent length from O to B.

Then

Pressure at $B = 405.70$ lb. per sq. in.
Flow B to $C = 45{,}000 + 349{,}570 = 394{,}570$ lb. per hr.

Drop B to $C = 1.55 \times 1.14 = 1.77$ lb. per sq. in.
Pressure at $C = 405.7 - 1.77 = 403.93$ lb. per sq. in.

Flow O to $P = 349{,}570 - 45{,}000 = 304{,}570$ lb. per hr.
Drop O to $P = 1.52 \times 0.70 = 1.07$ lb. per sq. in.
Pressure at $P = 405.80 - 1.07 = 404.73$ lb. per sq. in.

Now the pressure at P is greater than that at C and there must be a flow of steam from P to C
Pressure drop P to $C = 404.73 - 403.93 = 0.80$ lb. per sq. in.
Pressure drop per 100 ft. P to $C = 0.80 \div 2.07 = 0.39$ lb. per sq. in.
From Fig. 4: Flow from P to $C = 220{,}000$ lb. per hr.

Flow C to $D = 394{,}570 + 220{,}000 - 582{,}620 = 31{,}950$ lb. per hr.
Drop C to $D = 1.55 \times 0.01 = 0.02$ lb. per sq. in.
Pressure at $D = 403.93 - 0.02 = 403.91$ lb. per sq. in.

Flow D to $E = 31{,}950 + 349{,}570 = 381{,}520$ lb. per hr.
Drop D to $E = 1.55 \times 1.07 = 1.67$ lb. per sq. in.
Pressure at $E = 403.91 - 1.67 = 402.24$ lb. per sq. in.

Flow P to $Q = 304{,}570 - 220{,}000 = 84{,}570$ lb. per hr.
Drop P to $Q = 1.52 \times 0.07 = 0.11$ lb. per sq. in.

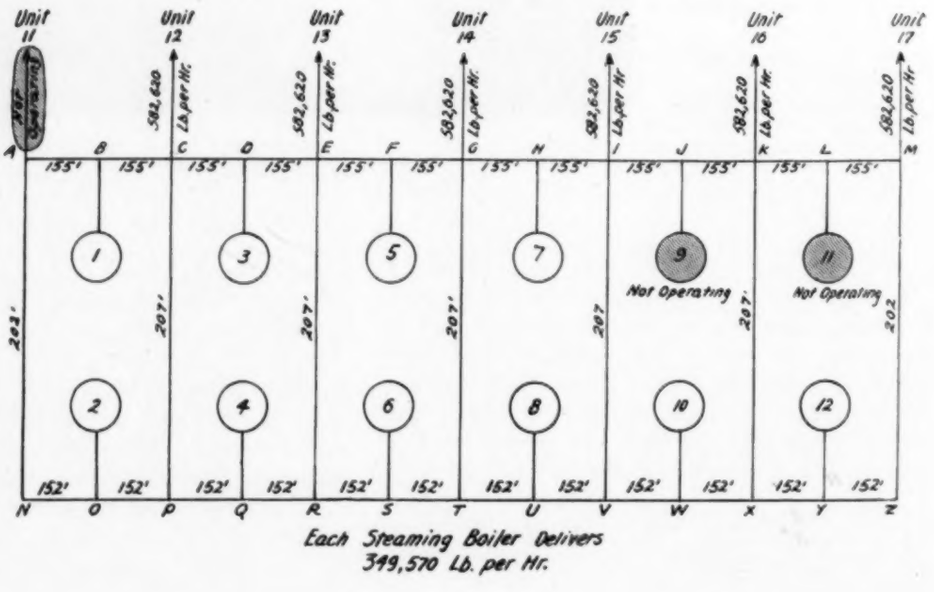


FIG. 3—SCHEMATIC DIAGRAM SHOWING TOTAL EQUIVALENT LENGTH BETWEEN INTERSECTIONS OF THE HEADER. WITH THIS DIAGRAM, CALCULATION OF PRESSURE LOSS CAN BE CARRIED OUT FOR ANY DESIRED CONDITION OF OPERATION

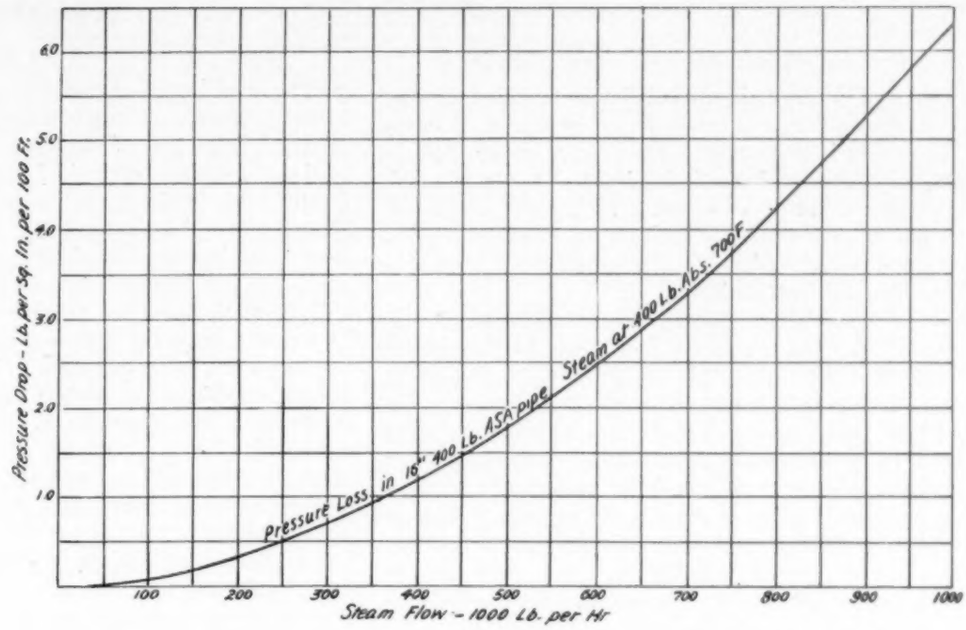


FIG. 4—IT WILL BE FOUND CONVENIENT TO PLOT A CURVE SHOWING THE RELATIONSHIP OF PRESSURE DROP TO WEIGHT OF STEAM FLOWING

Pressure at $Q = 404.73 - 0.11 = 404.62$ lb. per sq. in.

Flow Q to $R = 84,570 + 349,570 = 434,140$ lb. per hr.
Drop Q to $R = 1.52 \times 1.36 = 2.07$ lb. per sq. in.
Pressure at $R = 404.62 - 2.07 = 402.55$ lb. per sq. in.

Now the pressure at R is greater than that at E and there will be a flow of steam from R to E.
Drop R to $E = 402.55 - 402.24 = 0.31$ lb. per sq. in.
Drop per 100 ft. R to $E = 0.31 \div 2.07 = 0.15$ lb. per sq. in.
Flow R to E (from Fig. 4) = 131,000 lb. per hr.

Flow E to $F = 381,520 + 131,000 - 582,620 = -70,100$ lb. per hr.
Drop E to $F = 1.55 \times (-) 0.05 = (-) 0.08$ lb. per sq. in.
Pressure at $F = 402.24 - (-) 0.08 = 402.32$ lb. per sq. in.

Flow F to $G = 349,570 - 70,100 = 279,470$ lb. per hr.
Drop F to $G = 1.55 \times 0.60 = 0.93$ lb. per sq. in.
Pressure at $G = 402.32 - 0.93 = 401.39$ lb. per sq. in.

Flow R to $S = 434,140 - 131,000 = 303,140$ lb. per hr.
Drop R to $S = 1.52 \times 0.70 = 1.06$ lb. per sq. in.
Pressure at $S = 402.55 - 1.06 = 401.49$ lb. per sq. in.

TABLE 2

SECTION	FLOW	LENGTH	PRESSURE DROP	PRESSURE AT BEGINNING OF SECTION	PRESSURE AT END OF SECTION
O-B	45,000	509	5.09 × 0.02 = 0.10	405.80	405.70
B-C	394,570	155	1.55 × 1.14 = 1.77	405.70	403.93
O-P	304,570	152	1.52 × 0.70 = 1.07	405.80	404.73
P-C	220,000	207	2.07 × 0.39 = 0.80	404.73	403.93
C-D	31,950	155	1.55 × 0.01 = 0.02	403.93	403.91
D-E	381,520	155	1.55 × 1.07 = 1.67	403.91	402.24
P-Q	84,570	152	1.52 × 0.07 = 0.11	404.73	404.62
Q-R	434,140	152	1.52 × 1.36 = 2.07	404.62	402.55
R-E	131,000	207	2.07 × 0.15 = 0.31	402.55	402.24
E-F	− 70,100	155	1.55 × (−) 0.05 = −0.08	402.24	402.32
F-G	279,470	155	1.55 × 0.60 = 0.93	402.32	401.39
R-S	303,140	152	1.52 × 0.70 = 1.06	402.55	401.49
S-T	652,710	152	1.52 × 2.90 = 4.41	401.49	397.08
T-G	−546,000	207	2.07 × (−) 2.08 = −4.31	397.08	401.39
G-H	−849,150	155	.....	.....	.....
H-I	−499,580	155	.....	.....	.....

Flow S to $T = 303,140 + 349,570 = 652,710$ lb. per hr.
Drop S to $T = 1.52 \times 2.90 = 4.41$ lb. per sq. in.
Pressure at $T = 401.49 - 4.41 = 397.08$ lb. per sq. in.

Now the pressure at G is greater than that at T and there will be a flow of steam from G to T
Drop from G to $T = 401.39 - 397.08 = 4.31$ lb. per sq. in.
Drop per 100 ft. G to $T = 4.31 \div 2.07 = 2.08$ lb. per sq. in.
Flow G to T (from Fig. 4) = 546,000 lb. per hr.

TABLE 3

SECTION	FLOW	LENGTH	PRESSURE DROP	PRESSURE AT BEGINNING OF SECTION	PRESSURE AT END OF SECTION
O-B	55,000	509	5.09 × 0.03 = 0.15	405.80	405.65
B-C	404,570	155	1.55 × 1.20 = 1.86	405.65	403.79
O-P	294,570	152	1.52 × 0.66 = 1.01	405.80	404.79
P-C	248,000	207	2.07 × 0.48 = 1.00	404.79	403.79
C-D	69,950	155	1.55 × 0.05 = 0.07	403.79	403.72
D-E	419,520	155	1.55 × 1.28 = 1.99	403.72	401.73
P-Q	46,570	152	1.52 × 0.02 = 0.03	404.79	404.76
Q-R	396,140	152	1.52 × 1.15 = 1.75	404.76	403.01
R-E	283,000	207	2.07 × 0.62 = 1.28	403.01	401.73
E-F	119,900	155	1.55 × 0.13 = 0.20	401.73	401.53
F-G	469,470	155	1.55 × 1.58 = 2.45	401.53	399.08
R-S	113,140	152	1.52 × 0.11 = 0.17	403.01	402.84
S-T	462,710	152	1.52 × 1.53 = 2.33	402.84	400.51
T-G	301,000	207	2.07 × 0.69 = 1.43	400.51	399.08
G-H	187,850	155	1.55 × 0.29 = 0.44	399.08	398.64
H-I	537,420	155	1.55 × 2.02 = 3.14	398.64	395.50
T-U	161,710	152	1.52 × 0.22 = 0.33	400.51	400.18
U-V	511,280	152	1.52 × 1.84 = 2.80	400.18	397.38
V-I	348,000	207	2.07 × 0.91 = 1.88	397.38	395.50
I-J	302,800	155	1.55 × 0.70 = 1.08	395.50	394.42
J-K	302,800	155	1.55 × 0.70 = 1.08	394.42	393.34
V-W	163,280	152	1.52 × 0.22 = 0.34	397.38	397.04
W-X	512,850	152	1.52 × 1.85 = 2.82	397.04	395.22
X-K	348,000	207	2.07 × 0.91 = 1.88	395.22	393.34
K-L	68,180	155	1.55 × 0.04 = 0.07	393.34	393.27
L-M	68,180	155	1.55 × 0.04 = 0.07	393.27	393.20
X-Y	164,850	152	1.52 × 0.23 = 0.35	395.22	394.87
Y-Z	514,420	152	1.52 × 1.86 = 2.83	394.87	392.04
Z-M	514,420	202	2.02 × 1.86 = 3.76	392.04	388.28

Flow G to $H = 279{,}470 - 582{,}620 - 546{,}000 = -849{,}150$ lb. per hr.

Flow H to $I = 349{,}570 - 849{,}150 = -499{,}580$ lb. per hr.

Table of Trial Calculations: It is now evident that the original assumption is in error, for recirculation of steam in this section is impossible. The results so far indicate steam going around in a circle, from G to T, T to V, then up one or more cross-overs V-I, X-K, and Z-M, and back from I to G. The above calculations are recorded in Table 2, which is a convenient form for use in making the trial calculations.

Table 3 presents the results of assuming 55,000 lb. per hr. flow from O to B and 294,570 lb. per hr. from O to P. The final pressures at M are 393.20 lb. when going around the front header and 388.28 lb. when going around the back header. This is as near being the same as the final pressures can be calculated without working out pressure drops to the third decimal place. In view of the assumed equivalent lengths of the fittings such an agreement should be considered satisfactory and the final pressure at M would probably be about 390.7 lb. per sq. in., making the total drop in the header 405.8—390.7 = 15.1 lb. per sq. in.

Similar solutions can be made for other operating conditions, it being desirable to select conditions which are as severe as possible and which at the same time are probable. In this way it is comparatively easy to determine whether a given lay-out will be suitable from the standpoint of pressure drop.

APPENDIX

Drop in Tee Between Boiler Leads and Header

Since erection of the first section of Delray P. H. 3 some pressure drop tests have been made on the main steam piping. One of the most interesting developments of these tests is a method of determining pressure drop through the 14 in. x 16 in. x 16 in. tee between the boiler leads and main header. This method is developed along lines indicated by similar tests with water made at the *Hydraulic Institute of the University of München.*[3]

This method relates the pressure drop to the ratio of flows through the different branches and the velocity head at one of the openings. Fig. 5 shows this relation for the combinations of flow indicated. Data for other combinations of flow are not available to date. The velocity head at point B multiplied by the constant corresponding to the ratio of flows in A and B gives the pressure drop from A to B. Velocity head is equal to $V^2y \div 2g$

where V = velocity in ft. per sec.

y = density in lb. per cu. ft.

g = acceleration due to gravity in ft. per sec. per sec.

[3] Described in a bulletin, "Mitteilungen des Hydraulischen Instituts der Technischen Hochschule München, Heft I and II by D. Thoma Dr. Ing., O. Professor." Described in part in *A. S. M. E.* publication "Hydraulic Laboratory Practice," 1929, p. 140.

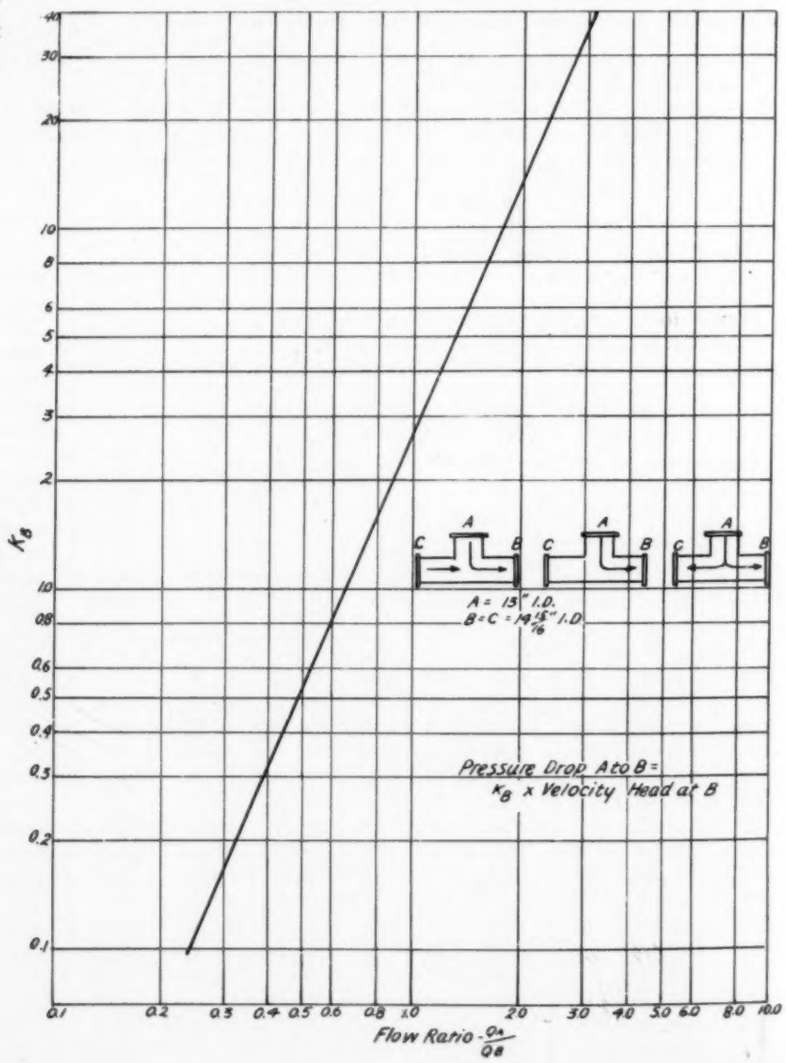


FIG. 5—CURVE SHOWING RELATION OF PRESSURE DROP IN TEE TO RATIO OF FLOWS THROUGH THE DIFFERENT BRANCHES

If similar data were available for each type of fitting used in the lay-out it would be possible to calculate the pressure drop over the system with the assurance of arriving at a correct answer. Such a calculation would be very laborious, however, since it would require the separate calculation of pressure drop through each fitting for each trial.

Since these data are not available for all types of fittings, and in view of the wide variety of operating conditions possible, it is not believed that the use of such refinement in the calculation of pressure drop is warranted.

Pressure Drop in Pipe Lines

A rational method of determining pressure drop in pipe lines of any size and carrying any kind of fluid has been developed in the last few years. This method, as applied to the flow of air and steam, is discussed by W. H. McAdams and T. K. Sherwood in *Mechanical Engineering* for October, 1926, pages 1025 to 1029. In this method, the pressure loss is assumed to vary as the square of the flow, and inversely as the diameter to the fifth power. The coefficient of friction is a function of the dimensionless factor $DVS \div Z$ where D is pipe diameter, V is velocity, S is specific gravity and Z is absolute viscosity.[4] This method is easily applied to a single line, but is difficult to use in the case of loop or divided circuits where the quantities flowing through various sections are not known at the start. Then, too, the answer obtained by this method is dependent upon the length of the line and is subject to the same errors in equivalent length as any other method. The rational method possesses one distinct advantage, however, in that available data for one fluid may be applied to any other fluid by proper use of the $DVS \div Z$ relation between the two fluids.

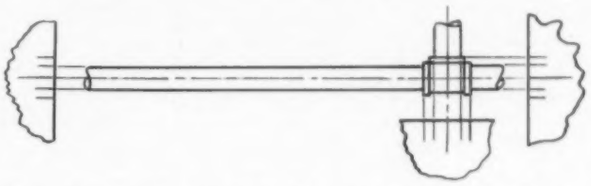
[4] Values of the terms used in the rational formula, expressed in convenient engineering units, are presented in the "Piping Handbook," by Walker and Crocker, p. 84-105.

Templet Aids in Detailing Piping

In making drawings of details of piping, in order to speed the work and improve appearance, the templet shown herewith was devised.

Its method of application is as follows:

When the center-line of the pipe has been drawn, the templet is placed across the center-line with the middle line on the templet for the pipe size in question over the center-line on the drawing. The two lines on the templet nearest to the center-line locate the outside of the pipe.

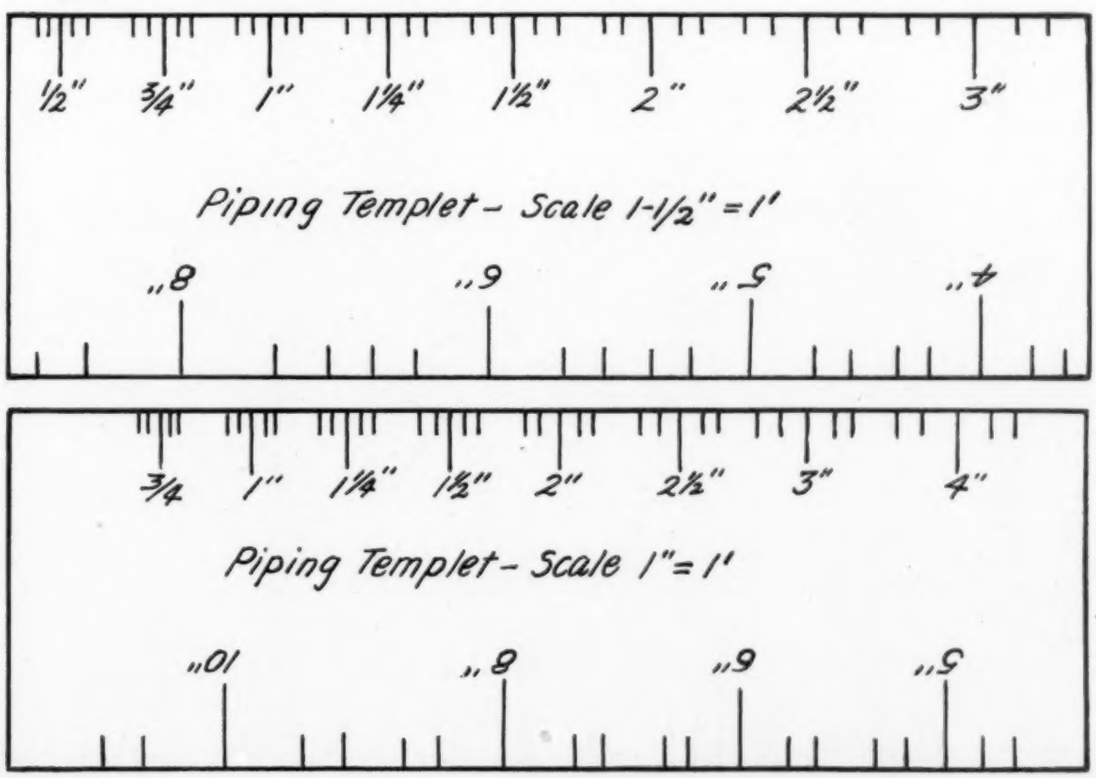


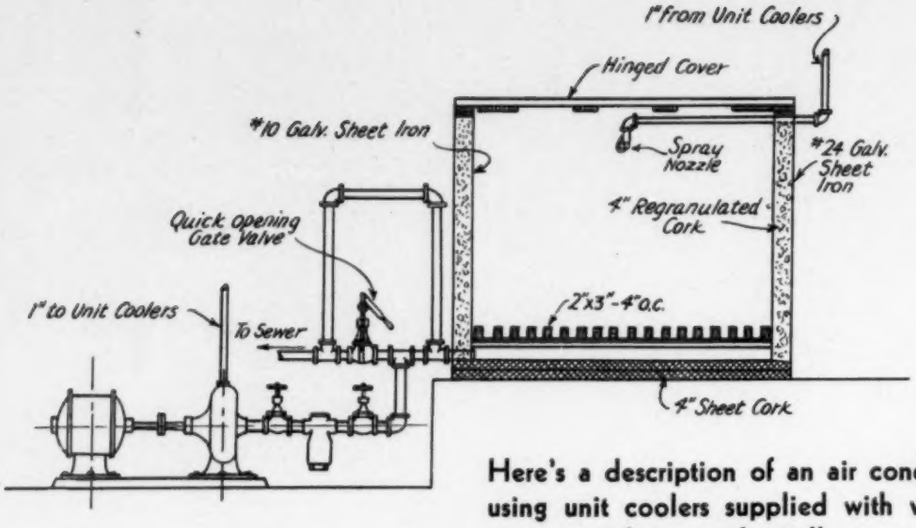
This Drawing Illustrates How the Templet Is Used in Detailing Piping

Marks are made accordingly on the drawing and the two parallel lines for the pipe are drawn.

When locating an elbow, tee or cross, the other two marks on the templet are used. The center-line is placed in the intersection of the center-lines where the fitting comes and the widest space on the templet locates the end of the elbow, tee or cross—*Clinton C. Hubbell.*

Templets for Assisting the Detailing of Piping May Be Constructed for Any Desired Scale; Two of Them Can Be Mounted Back to Back to Facilitate Use





WATER COOLING TANK FOR ICE COOLED AIR CONDITIONING SYSTEM

Here's a description of an air conditioning system in an office using unit coolers supplied with water cooled by ice. Information on the cost of installation, amount of power used, conditions maintained and details of the installation are given.

Office Conditioned Economically With Unit Coolers

—System is Cooled by Ice

LAST July an air conditioning system was installed in an office of the Knickerbocker Ice Company at Freeport, N. Y., for about $900.00, erected. The system is ice-cooled, consisting of two unit coolers, an ice-melting bunker and a 30-g.p.m. circulating pump.

The following tabulation of test temperatures shows the results that were obtained with this apparatus:

	Outside Temperature			
	D. B.	*W. B.*	*Eff. Temp.*	*R. H.*
July 29, 1931— 3 p. m.	83	74	78.5	66%
July 30, 1931—11 a. m.	83	78	80.5	80%
	Inside Temperature			
	D. B.	*W. B.*	*Eff. Temp.*	*R. H.*
July 29, 1931— 3 p. m.	75	64	70.5	54%
July 30, 1931—11 a. m.	73	63	69.5	57%

The cooled space totals approximately 10,500 cu. ft., the average occupancy being 8 adults. Frequently, 10 to 20 people were present in the office, and these overloads were satisfactorily handled by the cooling apparatus.

The power cost incident to the operation of the pump and fan motors as evidenced by the increase in the electric bills noted since the installation of the cooling system, averages $10.00 per month. The amount of power used apparently averaged 160 kw-hr. per month for August and for the portion of July during which the system was in operation.

The cooling water was taken from the bunker at 40 F, and returned at 42 F after passing through the units, there being 27 gallons, or 224 pounds of water, circulated per minute. No make-up water is used in this system as meltage from the ice furnishes more than enough water for the circulation system.

The refrigeration performed during normal operation was 448 Btu per min., or 2.24 tons for 24 hours. Actually, this amount of refrigeration was not required except for periods of short duration, even during extreme weather. The maximum ice consumption was 0.9 ton per 12-hr. operating day, the average for the summer being 0.5 ton per day.

The market value of the ice used is rather difficult to compute. Ice for retail use would average about $12.00 per ton delivered; small commercial customers would, in general, purchase ice for about $7.00 a ton. Special rates have, however, been made by most companies for ice used for air conditioning, and these rates run down to $4.00 per ton in many cases. For the type of installation described, $5.00 a ton could be considered a fair figure the country over, which would bring the net ice cost per average day to $2.50.

The unit coolers have an effective cooling surface of

By Clifford F. Holske*

75.3 sq. ft. each, according to the manufacturer's rating. The fan motors are equipped with speed controls, the tests being based on operation at the lowest speed, 700 r.p.m. At this speed, 370 c.f.m. of air passes through each cooler.

No direct provision has been made for the use of a predetermined amount of fresh air in this installation; the only outside air admitted is through one window which is maintained partially open, the exact opening depending on the requirements.

The average difference between the wet bulb temperature in the room and the cooling water temperature during the tests was 23 F. The total cooling surface being 150.6 sq. ft. the heat transfer was evidently in the neighborhood of 8 Btu per sq. ft. per hr. per degree temperature difference. Of course, a higher figure is obtained if the temperature of the air in the cooler is taken, rather than the general temperature of air in the room, as a basis for the calculation. However, as the final room temperature is the basic consideration, and as the temperature of the air in the cooler is a function of the room temperature, it would seem that one is justified in accepting the previous method for practical cooling system design, particularly in small systems where the portion of the load which is not susceptible to exact calculation represents a large percentage of the total load.

The fans on these units can be operated at 870 r.p.m. and 1,260 r.p.m. in addition to the lower speed previously mentioned. At these speeds the air handled is 460 and 667 c.f.m. respectively. Of course, the temperature drop is less at these higher capacities despite the increased heat transfer, due to the greater velocity. Evidently, it is well in figuring installations of this type to be liberal with the cooling surface, and to base the calculations of the capacity of the units upon the lowest fan speed. Not only does this provide for a reserve capacity in the equipment for handling unexpected overloads, but there is the additional advantage that the wet bulb temperature approaches the temperature of the cooling water more closely. This is particularly desirable where dehumidification of the air is of major importance.

* Test engineer, American Ice Company, New York.

A. S. T. M. Proceedings Published

Part I of the 1931 *A. S. T. M. Proceedings* contains the annual reports of the society committees and the papers and standards appended thereto. The president's annual address and the annual report of the executive committee are also included.

Part II of the *Proceedings* contains the technical papers which were given at the annual meeting of the Society in June, 1931.

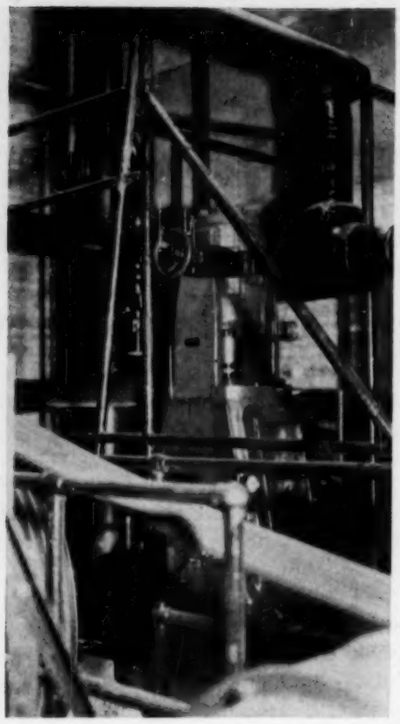
The subject matter of many of the papers each year can be grouped under the heading of metals. The 1931 *Proceedings* contains many papers on various aspects of fatigue, endurance testing, magnetic analysis, damping capacity and corrosion of metals. A symposium on malleable iron castings gives data on the properties of these types of castings.

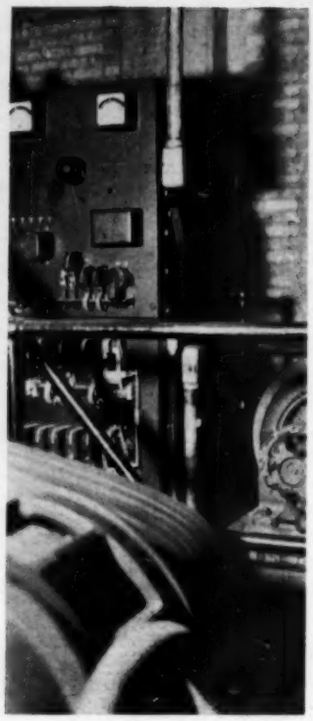
The economic significance of specifications for materials is a subject which is of importance to executives. It was discussed in a symposium held at the 1931 annual meeting; the papers and discussion comprise 44 pages of the *Proceedings*.

Copies of this publication are sold by the *American Society for Testing Materials,* 1315 Spruce st., Philadelphia.

THE AIR CONDITIONED OFFICE, SHOWING THE TWO UNIT COOLERS

By F. F. Sengstock*





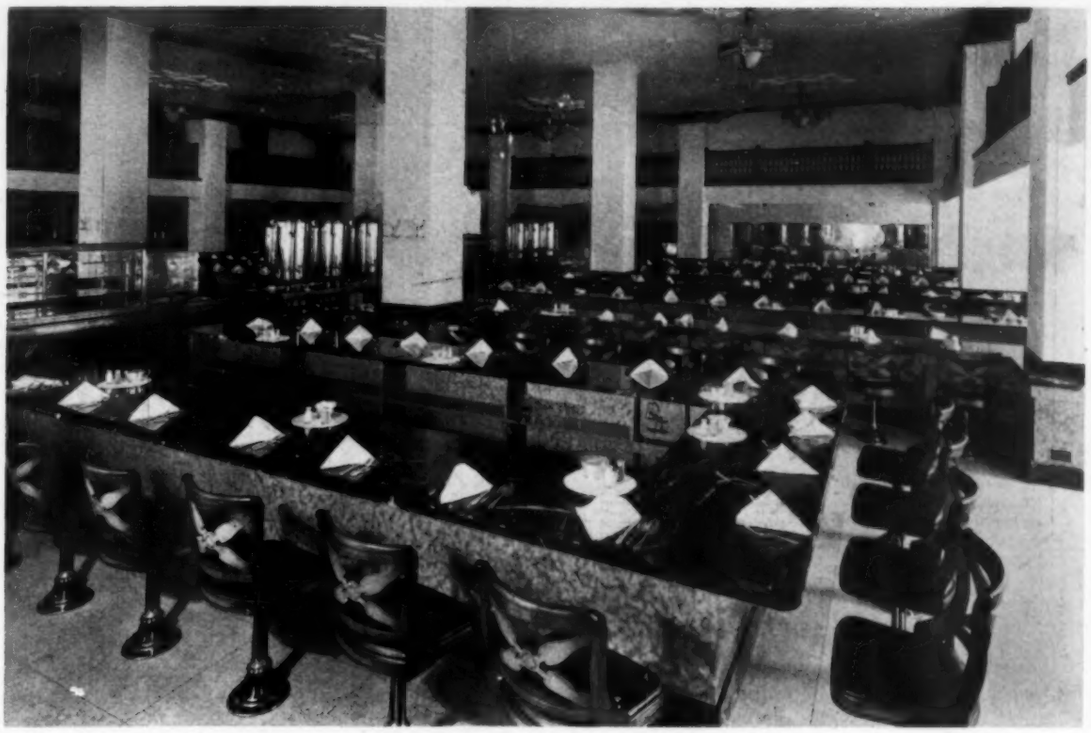
A View of the Refrigerating Equipment, Which Is Placed in an Elevator Pit Between the Car Hatchways

Compressors for Air Cooling Located in Elevator Pits

AFTER completion of the Merchandise Mart (Chicago), which has more than 4,600,000 sq. ft. of floor area and a volume of over 50,000,000 cu. ft. it was found necessary to install air conditioning equipment to supply certain areas, in order to insure the comfort of the tenants, their customers, and the employes. The purpose of the installation is to keep the air at 70 F in two of the areas on the first floor; one of these spaces is 80 ft. by 100 ft. with a ceiling height of 19 ft. The other is a large restaurant with several areas—80 ft. by 80 ft., one 60 ft.

* Formerly, General Superintendent, The Merchandise Mart, Chicago.

During the First Summer's Operation, It Was Found Advisable to Install an Air Cooling System for the Restaurant in the Merchandise Mart



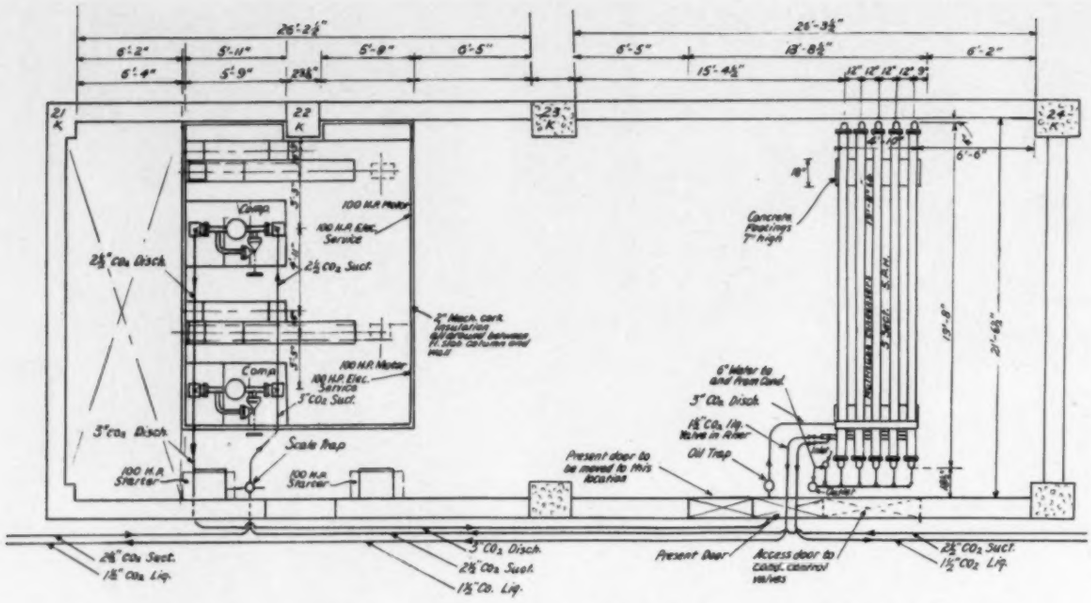
by 70 ft., another 120 ft. by 80 ft., in addition to the kitchen and typical occupied spaces of a large and well-managed restaurant.

TWO VIEWS OF THE RESTAURANT IN THE MERCHANDISE MART

Compressors Located in Elevator Pits

The problem of finding a location for the compressors and other equipment in an existing building is usually a difficult one. In this case it was solved, after some study, by placing this equipment in the elevator pits between the car hatchways.

The two CO_2 machines are of the single cylinder, double acting type and have a 6-in. bore and 6-in. stroke. They are driven by 100 h.p. synchronous motors, 440 volt, 3 phase, 60 cycles, with attached exciters. They are 7 ft. high over-all, and have a 66-in. flywheel, grooved for six cog belts. The two compressors, with appurtenances, occupy a space 17 ft. by 13 ft. and the coils require 18 ft. by 7 ft.

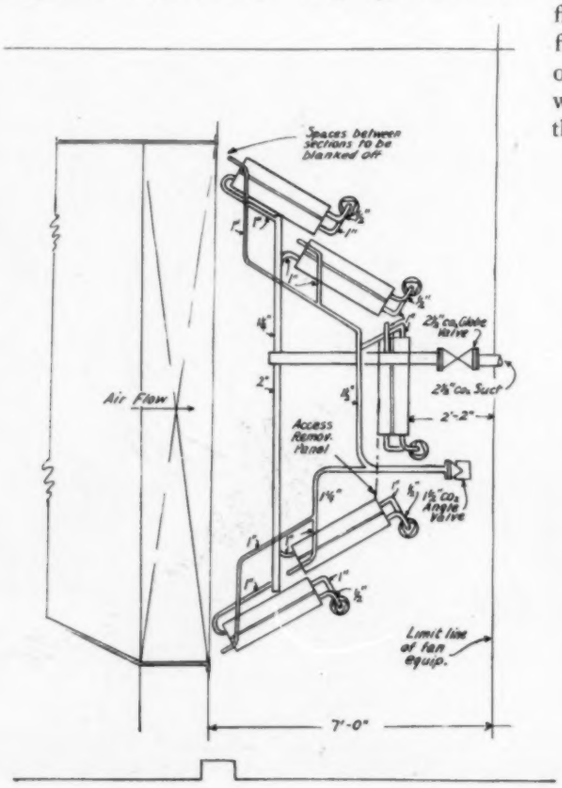


PLAN SHOWING LAYOUT OF THE REFRIGERATION MACHINERY IN THE ELEVATOR PIT

The compressors are located in one pit and the adjoining one is utilized for the coils and circulating system, besides providing room for carbon dioxide cylinders.

Cooling Coils in Ducts

The circulating water, at city pressure, is tapped in on the city main to the west of the Mart and runs about 200 ft. under a loading platform (the refrigerating equipment is located on the "track level" of the building). The return line connects with the line running back to the receiver tanks in the boiler room, about 200 ft. from the elevator pit and 40 ft. below the refrigerating equipment. The cooling coils are inserted in the air ducts approximately 100 ft. from the cooling equipment.



PLAN OF THE COOLING COILS AND CO_2 PIPING IN THE DUCT

Joints Welded

Pipe joints in the compressor room are welded, except for a few which are flanged. Supporting uprights are welded to the pipe lines to prevent vibration. Little serious difficulty was found in installing the piping, because of deep furring and plenty of available space on the track-level ceiling.

It is always a difficult matter to install air conditioning equipment in a completed building, however. For instance, due to the natural suction of the elevator shaft, dust from the necessary concrete cutting operations was carried as high as the fourth floor of the building. This was finally prevented through the use of wet burlap bags, kept soaked with running water, to dampen the dust.

Selecting *the Right Size* Heating Boiler

By Sabin Crocker*

Selection of the right size boiler is essential to the success of the installation. The information published herein will be found of great value, as the author has had wide experience in the problem.

This article is to be published in three consecutive installments—this is the first. In it, the author has gathered from a number of sources all of the information which is needed to select the proper size of steam or hot water heating boiler.

In this issue, the effect of amount of connected radiation, heat loss from the piping, and domestic hot water requirements on the required capacity of the boiler is considered.

Among the topics to be discussed in the second and third parts of this paper are warming-up period, efficiency with various fuels, grate area, combustion space, and type of heat liberation.

SELECTION of the right size steam or hot water heating boiler frequently is a vexing problem due to the number of factors involved and conflicting practices in rating such boilers. Among the items affecting size selection are:

(*1*) Amount of connected (standing) radiation.
(*2*) Heat loss from bare and covered piping in supply and returns.
(*3*) Domestic hot water requirements.
(*4*) Satisfactory warming-up period from a cold or partially cold condition of the entire system.
(*5*) Efficiency with hard or soft coal, gas- or oil-firing as the case may be.
(*6*) Grate area with hand-fired coal, or fuel burning rate with stokers, oil or gas.
(*7*) Combustion space in the furnace.
(*8*) Type of heat liberation, whether continuous or intermittent or a combination of both.

All of the foregoing factors have a definite bearing on the size boiler which should be selected for a given job, and should be taken into account in determining the proper amount of boiler heating surface and the requisite rate of heat liberation in the furnace. Many of the points discussed are well established and quite generally accepted, but it is deemed that a review and assembly will be of value.

Absorbing Capacity of Boiler Surface

Depending on the size of the job, the maximum heat absorbing capacity of the boiler surface should be from

*Engineer, The Detroit Edison Company, Detroit, Mich.

1¼ to 2 or more times the heat emission rate corresponding to the summation of Items *1, 2* and *3* above, and the heat liberating rate in the furnace should be correspondingly greater, depending on the boiler efficiency; otherwise action will be sluggish. With an under-size steam boiler it even may be impossible to get up steam pressure from a cold start with forced firing, while with an under-size water boiler it may require several hours to raise the water temperature to the desired point. Conditions of this kind are far from satisfactory and can be avoided through a better understanding of the principles underlying design. While the factors involved are simple in themselves, the necessity for considering each and every one frequently is overlooked in selecting a boiler.

Manufacturers' Ratings

Considerable confusion has resulted from the practice of some boiler manufacturers of rating their boilers as good for so many square feet of equivalent connected radiation (represented by the summation of Items *1, 2* and *3*), while other manufacturers have based their ratings on heat absorption in the boiler which is equivalent to including Item *4* with Items *1, 2* and *3*. Where boilers are bought on a competitive basis, there may be a tendency to skimp the size unless the person responsible for making a selection takes pains to analyze the heating surface and fuel burning capacity of the boilers offered by different bidders. Even where competition is not a factor, it frequently happens that under-size boilers are installed through failure to appreciate the true significance of manufacturers' ratings. Fortunately this tendency has been sensed by some boiler manufacturers who recently have included in their catalogs data on boiler heating surface, heat absorption corresponding to a given efficiency, boiler performance charts and the like. This is a big step forward and should do much toward correcting previous uncertainty regarding the true capacity of boilers. Such data are particularly essential at the present time in view of the different firing conditions obtaining with the variety of the solid, liquid and gaseous fuels now in common use.

The bearing which the items listed above have on the selection of an adequate boiler for any job can be brought out to best advantage by taking them up in detail one at a time.

Item 1—Computation of Load Due to Connected (Standing) Radiation

The commonly accepted basis for computing radiation load is the nominal square foot of connected or standing steam radiation. Different types of radiation (such as column, wall, pipe coil, bare pipe, indirect, etc.) have different heat emissions per square foot of actual surface. It is necessary, therefore, to reduce all surface to a common nominal basis, which for steam is accepted in the trade as one square foot of column (or tube) radiation containing steam at 215 F and in a room with an air temperature of 70 F. The heat emission of a nominal square foot of column steam radiation under the above conditions is defined as 240 Btu per hour, which corresponds to a condensation factor of approximately ¼ lb. per hr. The heat emission of a square foot of hot water radiation depends, of course, on the average water temperature maintained, which varies to a considerable extent in different types of open and closed systems. A frequently used value for heat emis-

TABLE 1—APPROXIMATE HEAT EMISSIONS, CONDENSATION FACTORS AND EQUIVALENT RATIOS FOR VARIOUS TYPES OF RADIATING SURFACES WITH ROOM TEMPERATURE OF 70 F[a]

Fluid	Radiation	Heat Emission Btu Per Sq. Ft. Per Hr.	Equivalent Sq. Ft. Column or Tube Radiation		Sq. Ft. Equivalent to One Sq. Ft. Column Steam	Condensation Factor, Lb. Per Sq. Ft. Per Hr.
			Steam[b] 240 Btu	Hot Water 150 Btu		
Steam at 215F..	Column or tube..	240	1.00	1.60	1.00	0.25
	Wall or pipe coil..	300	1.25	2.00	0.80	0.31
	Bare pipe.....	360	1.50	2.40	0.67	0.38
Water at 170 F..	Column or tube..	150	0.62	1.00	1.60	.....
	Wall or pipe coil..	190	0.80	1.27	1.26	.....
	Bare pipe.....	225	0.94	1.50	1.07	.....

[a] These are average values which do not take into account the actual number or height of columns, tubes or coils in a radiator nor whether the surface is horizontal or vertical. For more complete data, reference may be made to the *A. S. H. V. E. Guide*, particularly the 1928 edition in the case of column radiators.

[b] This column also represents heat emission relative to column steam radiation.

TABLE 2—CONVERSION FACTORS FOR COMPUTING UNDER GIVEN CONDITIONS (A) HEAT EMITTED BY COLUMN OR TUBE RADIATION, (B) EQUIVALENT NOMINAL RADIATION[a] [b]

Steam Pressure		Temp. of Steam or Water in F	Temperature of Air					
Vacuum In. Hg.	Lb. Absolute		80 F	75 F	70 F	65 F	60 F	50 F
22.4	3.72	150	0.388	0.424	0.462	0.499	0.538	0.617
20.3	4.74	160	0.462	0.499	0.538	0.577	0.617	0.697
17.8	5.99	170	0.538	0.577	0.617[d]	0.657	0.697	0.782
14.7	7.51	180	0.617	0.657	0.697	0.740	0.782	0.868
10.9	9.34	190	0.697	0.740	0.782	0.825	0.868	0.955
6.5	11.52	200	0.782	0.825	0.868	0.911	0.955	1.045
3.4	13.03	206	0.832	0.876	0.920	0.967	1.010	1.101
Lb.Gage[c]								
0	14.7	212	0.873	0.928	0.974	1.017	1.063	1.126
1	15.6	215	0.911	0.955	1.000	1.045	1.091	1.183
2	16.9	219	0.947	0.991	1.036	1.081	1.128	1.220
3	17.9	222	0.973	1.018	1.063	1.109	1.155	1.249
4	18.9	225	1.000	1.045	1.091	1.137	1.183	1.277
5	19.6	227	1.018	1.063	1.109	1.155	1.202	1.296
6	20.8	230	1.045	1.091	1.137	1.183	1.230	1.325
8	22.8	235	1.091	1.137	1.183	1.230	1.277	1.373
10	25.0	240	1.137	1.183	1.230	1.277	1.325	1.421

[a] Based on conversion factor of *A. S. H. V. E. Guide* equal to $[(tf-ta)/(215-70)]^{1.8}$ where tf = fluid temperature, ta = air temperature.

[b] (Conversion factor) x (heat emission of conductor containing 215-F fluid in air at 70 F) = (Btu per hr. heat emission of one sq. ft. of radiation under given conditions).

(Conversion factor) x (sq. ft. of conductor operating under given conditions) = (equivalent sq. ft. of same conductor containing 215-F fluid in air at 70 F).

[c] Approximate.

[d] This factor based on the heat emission ratio of nominal hot water radiation to nominal steam radiation would be 150 ÷ 240 = 0.625.

sion of hot water column radiation is 150 Btu per sq. ft. per hr., on which basis it takes 1.6 sq. ft. of hot water radiation to equal one sq. ft. of steam. This corresponds to an average water temperature of 170 F with a room temperature of 70 F. Values sometimes used for variations of these systems are 200-220 Btu per sq. ft. per hr. for vapor system radiators and 200-240 for accelerated hot water radiators. To avoid unnecessary duplication in attempting to distinguish between the various types of steam and hot water systems, this article will be confined in the main to steam heating with the understanding that the same general principles for the sizing boilers apply to the others except where otherwise pointed out.

Table 1 offers a basis for reducing different types of radiation to a nominal basis of one sq. ft. of column radiation in air at 70 F.

In case the radiation contains steam or water at temperatures other than 215 or 170 F, respectively, or the room temperature is other than 70 F, the conversion factors of Table 2 can be used as a basis for computing the equivalent nominal square feet. This table can be utilized in converting radiation of vapor or vacuum systems operating at temperatures other than 215 F into sq. ft. of 240-Btu radiation.

In the case of unit heaters, wall cabinets, and the like, the manufacturers' ratings should be used as a basis for computing the equivalent nominal radiation. After all installed radiation has been converted to its equivalent sq. ft. of nominal radiation, it should be summed up with the insulated and bare pipe losses and water heater load computed as described in subsequent paragraphs.

Item 2—Computation of Load Due to Bare and Covered Piping

Boiler manufacturers' catalogs frequently recommend adding some flat percentage allowance to the equivalent nominal radiation to care for heat loss from bare and covered pipe in the supply and return lines. Such percentage allowances commonly are given as 25 per cent of the equivalent nominal radiation for steam systems and 35 per cent for hot water, although allowances as low as 10 and 15 per cent, respectively, have been suggested, presumably where the piping was covered. Use of a flat percentage allowance is preferable to ignoring entirely the load due to heat loss from the supply and return lines, but better practice, especially where there is much bare pipe, calls for an actual computation of the emission from both bare and covered pipe surface, since actual heating systems differ considerably in this respect.

The amount of pipe left uncovered will vary in different jobs depending on whether conservation of fuel or other reasons dictate covering supply and return lines. In many cases where the supply and return lines pass through heated spaces it is advantageous to omit covering and install a corresponding lesser amount of standing radiation, thus saving both the cost of extra radiation and covering. It always is good practice to insulate pipes in separate boiler rooms, and usually in unheated basements, attics and similar places. In some instances, where large pipes pass through small rooms, insulation is required to prevent overheating the rooms. A further source of variation in the amount of surface in supply and return lines is the inherent difference between various types of systems such as one-or two-pipe steam or hot water. In general it is safe to say that more pipe surface exists in a two-pipe steam or water system than would be required for a corresponding one-pipe steam job. The actual heat loss will depend, of course, on the relative amounts of bare and covered pipe. Heating systems consisting mainly of indirect surface or large unit heaters will require much less piping than where column radiation of the same equivalent capacity is used.

After computing the actual amount of bare pipe surface existing in any heating system, due allowance must be made for the large heat emission of this type of surface. For approximate results, the factors shown in Tables 1 and 2 may be employed for conversion to a nominal radiation basis. Where more exact results are

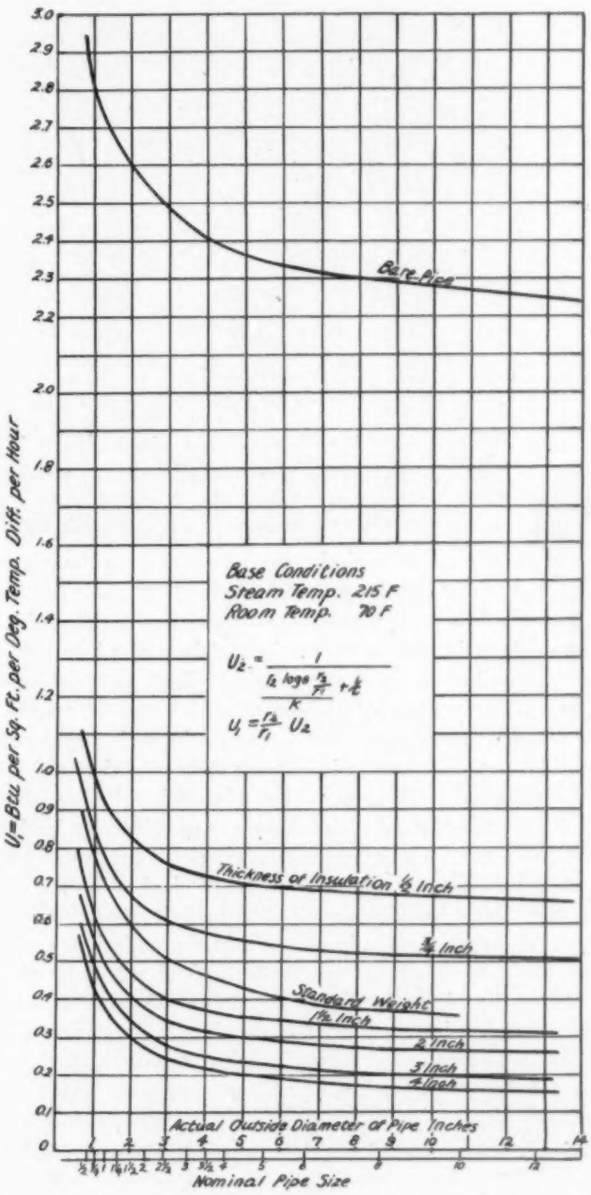


FIG. 1—VARIATION WITH PIPE SIZE OF RATE OF HEAT TRANSFER THROUGH VARIOUS THICKNESSES OF 85 PER CENT MAGNESIA INSULATION AND THROUGH BARE PIPE (See paper by L. B. McMillan, *Trans. A. S. M. E.*, Vol. 48, p. 1277)

TABLE 3—HEAT EMISSION FROM HORIZONTAL BARE IRON OR STEEL PIPES IN AIR AT 70 F

Nominal Pipe Size	External Surface		Heat Emission[a] Btu Per Linear Ft. Per Hr. Per F		Total Heat Emission[b] Btu Per Sq. Ft. Per Hr.		Equivalent Sq. Ft. of Column or Tube Radiation with Steam at 215 F		Equivalent Sq. Ft. of Column or Tube Radiation with Hot Water at 170 F		Condensation Factor, Lb. Per Hr. Per Sq. Ft. Steam at 215 F
	Linear Ft. Per Sq. Ft. Surface	Sq. Ft. Surface Per Linear Foot	Steam 215 F	Hot Water 170 F	Steam[c] 215 F	Hot Water 170 F	Steam 240 Btu	Hot Water 150 Btu	Steam 240 Btu	Hot Water 150 Btu	
⅛	4.547	0.220	0.643	0.594	425	270	1.76	2.82	1.13	1.80	0.441
¼	3.637	0.275	0.770	0.716	407	259	1.68	2.68	1.09	1.74	0.418
1	2.904	0.344	0.929	0.862	391	249	1.61	2.58	1.04	1.67	0.403
1¼	2.301	0.435	1.16	1.065	387	246	1.60	2.56	1.02	1.63	0.400
1½	2.010	0.498	1.30	1.197	379	241	1.57	2.51	1.00	1.60	0.392
2	1.608	0.622	1.58	1.461	368	234	1.53	2.45	0.98	1.57	0.383
2½	1.328	0.753	1.89	1.742	363	231	1.50	2.40	0.96	1.54	0.375
3	1.091	0.917	2.26	2.063	358	226	1.46	2.34	0.94	1.50	0.366
3½	0.954	1.047	2.53	2.333	350	223	1.45	2.31	0.92	1.48	0.362
4	0.848	1.178	2.81	2.600	346	220	1.43	2.28	0.92	1.47	0.356
5	0.686	1.456	3.41	3.140	340	216	1.40	2.24	0.90	1.44	0.350
6	0.576	1.734	4.05	3.703	338	213	1.40	2.24	0.89	1.42	0.349
8	0.443	2.257	5.19	4.720	334	209	1.39	2.23	0.87	1.39	0.348
10	0.355	2.817	6.38	5.820	329	207	1.37	2.19	0.86	1.38	0.343
12	0.299	3.338	7.53	6.870	327	206	1.36	2.18	0.86	1.37	0.340
18	0.212	4.717	10.41	9.50	320	202	1.33	2.13	0.84	1.35	0.332
Flat	.....	.....	.....	.....	303	192	1.26	2.02	0.80	1.28	0.315

[a] Based on experimental data obtained at the Mellon Institute.

[b] An average heat emission with steam at 215 F and air at 70 F is 360 Btu per sq. ft. per hr. and for hot water at 170 F and air at 70 F, 225 Btu per sq. ft. per hr.

[c] Conversion to other conditions of fluid or air temperatures may be accomplished by use of the following formula: $H = H_s\,[(tt\text{-}t_a)/(215\text{-}70)]^{1.25}$ where H = heat emitted under desired conditions, tt and t_a = corresponding fluid and air temperatures respectively, H_s = heat emitted with fluid temperature of 215 F in air at 70 F. Conversion factors given in Table 2 and computed from the similar formula $H = H_s\,[(tt\text{-}t_a)/(215\text{-}70)]^{1.3}$ will produce results sufficiently accurate for most practical applications.

desired, the amount of surface of each size of bare pipe should be computed and the corresponding heat emissions figured from data such as that shown in Table 3.

Reference to Table 3 indicates that small pipes have considerably larger emissions per square foot than large pipes, which may have a decided effect on the total amount of heat emitted from the bare pipe surface. Data given in Table 3 are for horizontal pipes. The corresponding losses from vertical pipes are somewhat less, but in the absence of specific values may be assumed as the same as for horizontal pipes. Attention to details of this kind frequently is important in securing a properly balanced system and size of heating boiler. Neglect to estimate bare pipe losses correctly may lead to overheating of the spaces through which the pipes pass and result in considerable waste of fuel. Where excessive bare pipe emission is indicated, it is advisable to cover enough of the pipe to reduce the emission to that required to heat the room to the desired temperature.

Heat loss from insulated pipes also should be taken into account in computing boiler loads. Where ¾-in. thick insulation is used, the heat loss can be assumed as approximately 25 per cent of that from the same size of bare pipe under the given conditions for fluid and air temperature. Where covering of greater or less thickness than ¾-in. is used, reference may be made to Fig. 1 giving heat transfer coefficients for bare pipe and for various thicknesses of covering on the different nominal pipe sizes.

It should be understood that the 25 per cent heat loss assumption is an approximation representing a compromise figure for different pipe sizes. Actually, a given thickness of covering is much less efficient on a small pipe than on a large one due to relatively greater increases in outer surface exposed to the air. This is compensated for in part if the heat loss from covered pipe is taken as 25 per cent of the bare pipe losses proportioned to nominal pipe diameter as indicated in Table 3. Where a more accurate figure is required, the loss through the corresponding thickness of insulation can be computed for the amount of pipe of each size in the system, using values from the curves of Fig. 1. According to the *A. S. H. V. E.* Code,[1] pipe covering less than ¾-in. should be considered the same as bare pipe in computing boiler load, while pipe covering ¾-in. or more in thickness is considered as having 25 per cent the heat loss of bare pipe. This is a rather rough approximation and the computation of heat loss in any given instance can be refined as much as circumstances seem to dictate.

That the variation between different systems is of considerable magnitude is evidenced by Table 4 which is made up from a few typical jobs for similar types of buildings coming within the writer's experience.

Table 4 is offered as an illustration of the extent to which the surface of supply and return lines may vary between different types of systems, rather than as a basis for setting definite percentages for design purposes.

[1] "Code for Rating Steam Heating Solid Fuel Boilers." (Adopted 1929, revised 1930.)

FIG. 2—EXTERIOR OF MT. OLIVET WAREHOUSE OF THE DETROIT EDISON COMPANY

FIG. 3—INTERIOR OF MT. OLIVET WAREHOUSE

Particular attention is called to the small amount of piping required for the unit heater jobs in the Ann Arbor and Mount Olivet warehouses included in Table 4.

TABLE 4—PIPING TAX FOR SEVERAL INDUSTRIAL BUILDINGS

All radiation and piping reduced to a common basis of 240-Btu nominal steam radiation.

	Standing Radiation Sq. Ft.	Piping Loss			% Piping Loss to Standing Radiation
		Bare	Covered[a]	Total	
Mt. Olivet substation, 2-pipe steam-column radiation	1432	30	151	181	12.6
Mt. Olivet warehouse, 2-pipe steam, floor type unit heaters	7500	56	171	227	3.0
Ann Arbor warehouse, 2-pipe steam, suspended type unit heaters	7660	600	43	643	8.4
Garfield substation, 2-pipe hot water column radiation	4285	1533	243	1776	41.5
Pulford substation, 1-pipe steam column radiation	4978	828	126	954	19.1
Frisbie substation, 2-pipe steam column radiation	2724	.....	273	273	10.0

[a] The loss from covered pipe has been assumed as 25% of the loss from the same size bare pipe.

To give an idea of the nature of these buildings, a photograph of the exterior of Mount Olivet warehouse is shown in Fig. 2 and an interior view—in which the two unit heaters appear—in Fig. 3. One heater is in the background about the middle of Fig. 3 and part of the other appears at the extreme left. Eventually the building will be extended and the heater in the middle of the picture moved farther along. An office and toilet room in one corner of the building above the boiler are heated with hot water radiators supplied from a submerged heater alongside the boiler. The hot water radiation with its piping is not included in the values of Table 4.

Item 3—Computation of Load Due to Domestic Hot Water Requirements

Where the domestic hot water supply is heated from a building heating boiler, the corresponding load must be taken into account in sizing the boiler. Btu required per hour and equivalent nominal radiation corresponding to various gallon-per-hour water heating rates are given in Table 5 for a series of temperature rises. Indirect heaters,[2] fire box coils and storage tanks sometimes are rated on a 3-hr. basis instead of a 1-hr., or the rated

[2] "Indirect heaters" as used here refers to copper coils or other surface placed below the boiler water line.

water temperature rise may be given as 1/3 of that required to give a satisfactory water temperature. Where this is the case, a corresponding adjustment should be made in using Table 5.

The water heating loads given in Table 5 represent average values over the entire period of heating plus an allowance of 15 per cent. The 15-per cent increased capacity provides an allowance to compensate for radiation losses from the storage tank and circulating piping after the water has begun to warm. The values given in Table 5 agree with the figure of 1 1/3 sq. ft. of steam radiation (or 2⅛ sq. ft. of water radiation) per gallon of storage tank capacity sometimes given in heating supply catalogs, bearing in mind that the 1 1/3 sq. ft. figure is on a basis of taking three hours to heat the tank to a satisfactory temperature. The value of ½ sq. ft. of steam radiation given in some heating supply catalogs is entirely inadequate if the storage tank is to be warmed in three hours. There is some justification, however, for the practice of omitting the application of any further piping tax or pick-up allowance for the water heating load beyond the 1 1/3 sq. ft. referred to above.

There are at least eight items which must be considered when selecting the right size heating boiler. While many of the points discussed in this article are well established and quite generally accepted, it is felt that this review and assembly will be found of great value to engineers.

Indirect water heaters are sold on a basis of their water heating capacity, and supply catalog data can be used as a basis for computing load on the boiler. Similar ratings are available for some types of direct water heaters known as fire pot generators, water backs and the like. In the case of pipe coil heaters placed in the fire box, a heat transfer rate of 12,000 to 15,000 Btu per sq. ft. per hr. can be assumed, depending on how the coil is placed with respect to the fire and the extent to which it has scaled up. This is equivalent to a 240-Btu radiation load of 50 to 62.5 sq. ft. per sq. ft. of coil surface. The same values can be applied in the case of cast iron or brass water backs where catalog information is not available. Assuming a 100-F water temperature rise, each sq. ft. of water back or fire box coil surface is capable of heating 12½ to 15½ gallons of water per hour, including a 15-per cent allowance for radiation losses from storage tank and circulating piping.

TABLE 5—WATER HEATING LOADS VS. TEMPERATURE RISE FOR VARIOUS GALLON-PER-HOUR RATES[a]

Gallons Heated Per Hour	100 F Rise		110 F Rise		120 F Rise		130 F Rise		140 F Rise		150 F Rise	
	1,000 Btu Per Hr.	Equivalent Sq. Ft. 240 Btu Radiation	1,000 Btu Per Hr.	Equivalent Sq. Ft. 240 Btu Radiation	1,000 Btu Per Hr.	Equivalent Sq. Ft. 240 Btu Radiation	1,000 Btu Per Hr.	Equivalent Sq. Ft. 240 Btu Radiation	1,000 Btu Per Hr.	Equivalent Sq. Ft. 240 Btu Radiation	1,000 Btu Per Hr.	Equivalent Sq. Ft. 240 Btu Radiation
5	4.80	20	5.28	22	5.76	24	6.24	26	6.72	28	7.20	30
10	9.60	40	10.56	44	11.52	48	12.48	52	13.44	56	14.40	60
20	19.20	80	21.12	88	23.04	96	24.96	104	26.88	112	28.80	120
30	28.80	120	31.68	132	34.56	144	37.44	156	40.32	168	43.20	180
40	38.40	160	42.24	176	46.08	192	49.92	208	53.76	224	57.60	240
50	48.00	200	52.80	220	57.60	240	62.40	260	67.20	280	72.00	300
60	57.60	240	63.36	264	69.12	288	74.88	312	80.64	336	86.40	360
70	67.20	280	73.92	308	80.64	336	87.36	364	94.08	392	100.8	420
80	76.80	320	84.48	352	92.16	384	99.84	416	107.5	448	115.2	480
90	86.40	360	95.04	396	103.7	432	112.3	468	121.0	504	129.6	540
100	96.00	400	105.6	440	115.2	480	124.8	520	134.4	560	144.0	600
120	115.2	480	126.7	528	138.2	576	149.8	624	161.3	672	172.8	720
140	134.4	560	147.8	616	161.3	672	174.7	728	188.2	784	201.6	840
160	153.6	640	169.0	704	184.3	768	199.7	832	215.0	896	230.4	960
180	172.8	720	190.1	792	207.4	864	224.6	936	241.9	1008	259.2	1080
200	192.0	800	211.2	880	230.4	960	249.6	1040	268.8	1120	288.0	1200
250	240.0	1000	264.0	1100	288.0	1200	312.0	1300	336.0	1400	360.0	1500
300	288.0	1200	316.8	1320	345.6	1440	374.4	1560	403.2	1680	432.0	1800
400	384.0	1600	422.4	1760	460.8	1920	499.2	2080	537.6	2240	576.0	2400
500	480.0	2000	528.0	2200	576.0	2400	624.0	2600	672.0	2800	720.0	3000

[a] Table is computed for water having a density of 8 1/3 lb. per gal. and a specific heat of one Btu per F. Loads are based on a boiler tax equal to 4 sq. ft. of 240-Btu steam radiation for each gallon of water heated per hour through 100 F, computed as follows:

Heat required to raise water temperature = 8 1/3 × 100 = 833 Btu/hr. = 3.47 sq. ft. steam radiation per gal.
Heat lost by radiation from storage tank and circulating piping = 127 Btu/hr. = 0.53

Total ..... 960 Btu/hr. 4.00

The allowance for radiation loss from storage tank and circulating piping represents approximately 15 per cent of the heat required to raise the water temperature.

In some installations, steam coils immersed in tanks are used for water heating purposes. Water heating capacities and corresponding boiler loads for such coils of brass or iron pipe can be read from Fig. 4. These values are for horizontal pipe coils and contain an allowance of 50 per cent for fouling of the pipes. Consequently the maximum load with clean, new pipe may be twice that shown, which should be taken into account in sizing the boiler. All pipe surfaces considered here and elsewhere in this article are external.

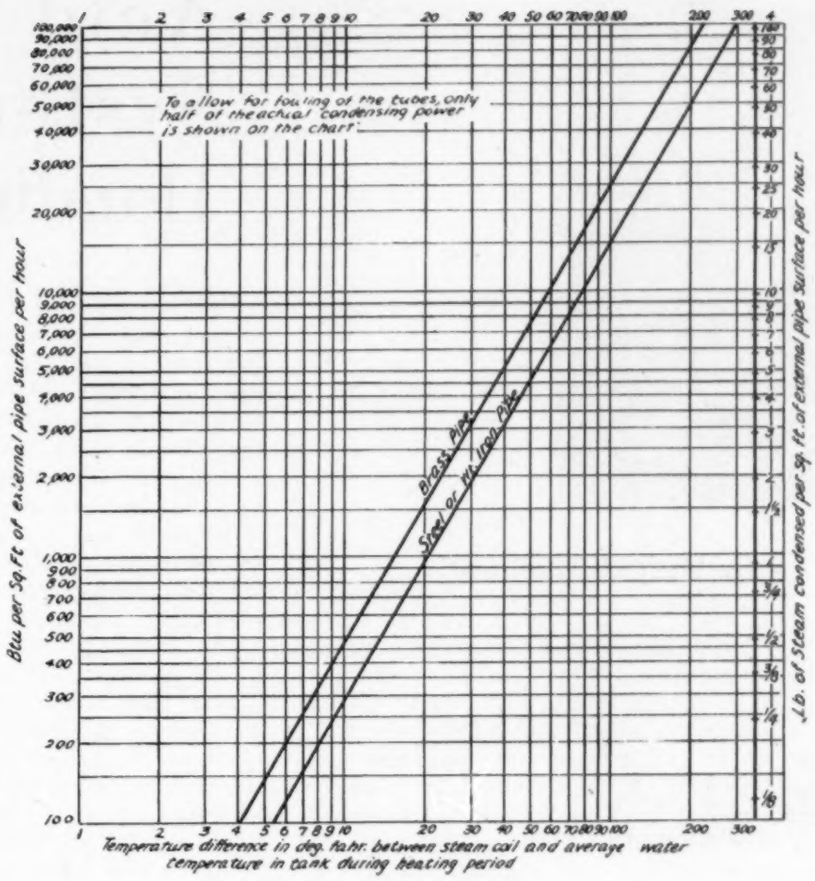
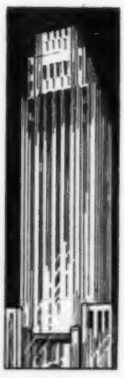

To allow for fouling of the tubes, only half of the actual condensing power is shown on the chart.
Brass Pipe
Steel or Wrought Pipe
Btu per Sq. Ft. of external pipe surface per hour
Lb. of Steam condensed per sq. ft. of external pipe surface per hour
Temperature difference in deg. fahr. between steam coil and average water temperature in tank during heating period

FIG. 4—HEATING POWER OF BRASS AND IRON PIPE COILS IN WATER STORAGE TANKS

(Based on tests made by the American Radiator Company.)



Cost of Cooling an Office Building

IN ONE large project, a 40-story office building recently analyzed, the heating system with mechanical exhaust for the toilet rooms cost for investment per square foot of rentable area 0.415 dollars.

Labor, fuel and power for this plant cost per square foot of rentable area.....0.064 dollars

If a complete ventilating system is added, it will cost for investment, in addition to the heating, per square foot of rentable area .....0.725 dollars

Operating cost for ventilating, per season, per square foot of rentable area.....0.023 dollars

The space occupied on each story by ventilating and cooling equipment is an interior one, behind the elevators, of little value for other purposes, and amounts to about 250 sq. ft. per story.

If, having installed the heating and ventilating systems as outlined, two duplicate refrigerating units are added (located in the basement) with pumps and cold water piping to the various air supply systems, the investment cost will be, per square foot of rentable area.....0.390 dollars

Operating cost for *cooling-for-comfort*, per season of 60 days per square foot of rentable area .....0.064 dollars

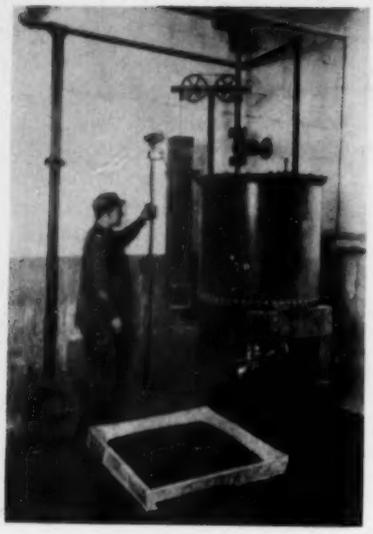
If 5 per cent interest and 2.72 per cent, twenty-year sinking fund on the investment cost is allowed and the operating cost is added we learn that the cost for investment and operation check out about as follows, per square foot of rentable area per season, in this building:

Heating	$0.0964	say ten cents
Ventilating	0.1290	say thirteen cents
Cooling-for-Comfort	0.0944	say ten cents

The ventilating system is necessary if it is desired to *cool-for-comfort.*

In other words if the modest sum of about 25 cents per square foot of rentable area on a big plant is added to the annual rental in this building *cooling-for-comfort* can be furnished.—*Samuel R. Lewis,* in a paper presented before the *Western Society of Engineers.*

Acid Piping System Saves Handling Costs



Sulphuric Acid Service Tank on Third Floor

By **Dwight Spofford***

AN ACID unloading and piping system that has proved to be economical in time, labor and cost of operation, is used in the plant of the Electro-Chemical Engraving Co., New York City.

Previous to the installation of the equipment, sulphuric and hydrochloric acids were delivered in carboys, necessitating truck-men, elevator usage to transfer the carboys to other floors, considerable storage space, and of course, requiring much time in handling. As only small quantities of acids were deliverable at a time, a higher price had to be paid as compared to the price in tank truck quantities.

James B. Monahan, chief engineer, conceived the idea of handling the acids by pumps and piping, and made the lay-out for the system. The sulphuric and hydrochloric acid handling systems are practically identical—plain steel tanks, pipe and acid-resisting valves for the sulphuric acid, and rubber-lined tanks, pipe and valves for the hydrochloric acid. Acids are now delivered by tank trucks.

A pipe tunnel was bored under the sidewalk to two 31-in. diameter cast iron man-holes four feet apart and 23-in. from the curb. Two 6-in. casings of tile pipe were laid through the tunnel. A 1½-in. extra heavy flanged steel pipe was run through for the sulphuric acid and a 2-in. rubber-lined steel pipe for the hydrochloric acid. The man-hole basins and fill caps are set in concrete.

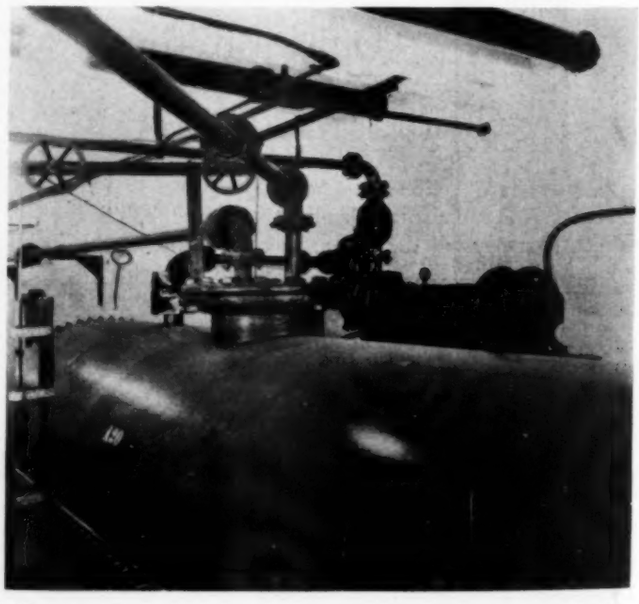
Main storage tanks are located in the basement of the seven-story building and are filled by siphoning from the tank truck. To start the siphon over the top of the tank truck, a pound or two of air is put on the tank by means of a small motor-operated pump. The acids are pumped from the main storage tanks to the third floor into 200-gal. tanks, from which they flow by gravity through pipes to the vats where they are to be used.

As the fire department has a regulation which prohibits leaving acid in any pipe lines, it was necessary to use self-priming pumps; they are acid-resisting, self-priming centrifugal pumps, direct connected to 5 hp., 1750 r.p.m., 2 phase, 60 cycle, 220 volt motors.

The main switch is in the basement in a room next to the storage tank room, with push button control nearby. Remote push button controls are located on the third floor next to the service tanks, with red lights showing when the pumps are operating. Floats in the tanks automatically cut off the pumps when the tanks are filled to within five or six inches of the top. To prevent siphoning back the inlet pipes extend only two inches below the tank covers. All acid remaining in the pipe lines after the pumps have been shut off drains back into the main storage tanks.

From 15 to 20 seconds after the "start" button is pressed on the third floor, acid is being delivered, taking but a few minutes to fill the tanks. As the tanks are filled twice a day, the time saved is a considerable item.

It has been estimated that the installation will pay for itself within three years by the difference in cost between acid in carboys, and in tank trucks, not counting savings in manual labor.



Main Storage Tanks in the Basement Are Filled by Siphoning From Tank Trucks

*The Duriron Co., Inc., Dayton, Ohio.

Piping Changes Made Without Harm to Service

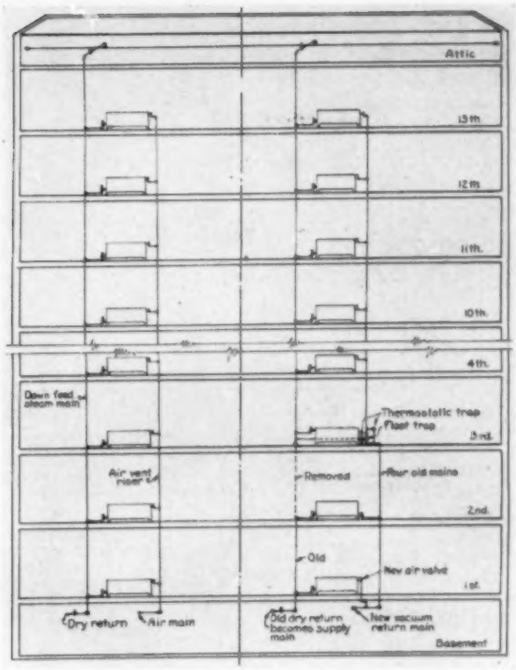


FIG. 1 (LEFT)—OLD OFFICE BUILDING AS IT WAS, WITH DOWN-FEED SINGLE-PIPE OR AIR-VENT SYSTEM OF STEAM HEATING

RIGHT—OLD OFFICE BUILDING AS REMODELED WITH TWO-PIPE VACUUM FOR THE LOWER THREE STORIES

How the Heating Plant in an Office Building Was Rejuvenated Without Shutting System Down

By Samuel R. Lewis*

THE steam and return piping in a large thirteen-story office building of the older type is shown in Fig. 1. It was heated by a down-feed steam system. Each radiator had a large single-pipe connection from the vertical main pitching up to the radiator, and steam and condensation flowed in opposite directions into and from the radiator. A ¼-in. air pipe with a pinhole orifice in each radiator at the end opposite the supply connection vented the radiator with a ¾-in. downgoing air exhaust main to a vacuum pump in the basement. The plant worked fairly well, but it became necessary to rearrange and rejuvenate the lower three stories.

One story only at a time could be vacated for reconstruction. The third story was chosen as the first one to be rebuilt, but the tenants on the first and second stories were temperamental and could not be disturbed. The third story could be commenced in summer, but could not be finished before cool weather, and inevitably the second story would be rebuilt during winter.

It was decided to install a two-pipe vacuum system. A new vacuum return main was extended along the walls of the building at the third story. The lower ends of four old down-feed risers through the first and second stories could be spared, and the new third story return mains were bled down through these and connected in the basement to a new electric vacuum pump of a size sufficient to serve the entire building.

The 65 old down-feed mains were cut and reconnected at the third story and were drained through float traps into the new vacuum return mains at the third story.

At the same time, in the basement, where work could be done without trouble with the tenants, the old dry return mains were fed with steam at six strategic points so as to avoid regrading them and so as to feed them often enough that they would have sufficient capacity to up-feed two stories.

A new vacuum return main also was run in the basement while the third-story work was being prosecuted, which picked up the four drop connections from the new third story vacuum returns.

It was thus possible to down-feed the upper parts of the building with only four of the lower ends of the old risers through the second story to serve as vacuum return mains. The first and second stories were up-fed from the old dry return mains which became supply mains.

* Consulting engineer, Chicago, Ill.

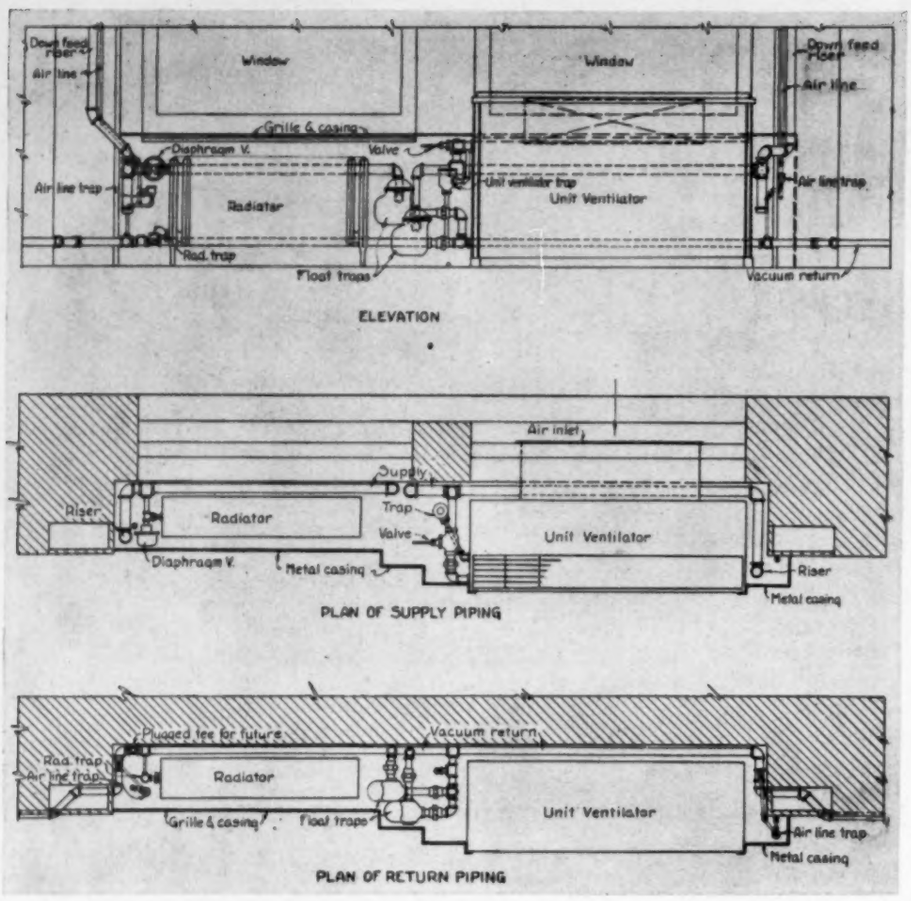


FIG. 2—ELEVATION AND PLANS OF PIPING AND TRAPS, WITH NEW RADIATORS AND UNIT VENTILATING MACHINES ON THE THIRD STORY

When the third story was completed the second-story tenants moved up there, and the second story was then rebuilt, removing entirely 61 of the old risers through that story and having ample heat during construction by steam up-fed from the old dry return main.

In general the second story as rebuilt has its own local return mains run behind the radiators and unit ventilating machines along the outer walls. These mains, since they serve but one story, are of course much smaller than those on the third story which serve ten stories.

It will be observed, if this story has been told clearly, that, so far, first-story tenants have not been disturbed.

It was thought that the orifice-type air vents might interfere with the successful maintenance of sub-atmospheric pressures by the new vacuum pump, as steam could conceivably pass them. Therefore, when the old air line mains were connected into the new vacuum mains at the third story, thermostatic traps which would pass water and air but which would not pass steam were installed. Where the old first-story radiators continued on the single-pipe system, air-line thermostatic air valves were utilized to supplant the pin-hole type air valves.

Fig. 2 shows a typical elevation at the third story with the unit ventilating machines, radiators, float traps, thermostatic traps, steam valves, etc. Fig. 3 indicates the work which some one performed in an attempt to seal the warring different dry return mains against each other.

It is believed that whenever the siphon at the right was primed by accumulation of condensation it must have siphoned itself dry, eliminating for a time the seal. The idea of the deeper water seal at the left was that it would act as a safety valve and waste to the sewer in the event that the condensation could not all return to the hot well. The peculiar return header shown in Fig. 3 was of course, eliminated when the vacuum pump was installed.

An interesting side light to the old piping was that occasionally the reciprocating engines of the power plant delivered more exhaust steam than the heating system could condense. There was a 12-in. single-disc back-presssure valve on the out-board exhaust pipe, which was not steam-tight. To prevent the noise made by this valve and to reduce wasteful leakage it was the custom to hang a mass of scale-weights on the operating arm. Thus the valve became inoperative and when undue back pressure did occur the water seals or siphons shown in Fig. 3 acted as relief valves and blew out steam after coughing themselves clear of water.

This back-pressure valve trouble was cured by re-grinding the big valve and by weighting it to hold five pounds pressure. A 6-in. multiport back pressure valve was installed and set to discharge at 3½ lb. on a by-pass around the big valve. The smaller valve with the many spring-loaded ports does almost all of the necessary work, with its big neighbor merely resting comfortably in the background as a reserve.

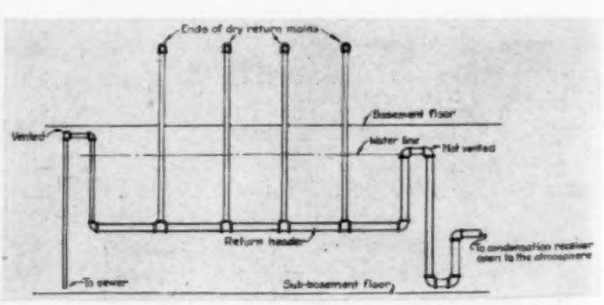


FIG. 3—OLD RETURN HEADER IN SUB-BASEMENT

The ends of the old dry return mains were sealed each against the other by the false water line arrangement shown. The 65 dry mains from over-head, however, were not sealed against each other.

Thirteen miles of 1¼-in. piping were used in the construction of the new ice skating rink at the University of Illinois, which was constructed by the athletic department for the use of the students. At the present time it is planned to use the rink for skating only, but blank tees are provided in order that provisions for circulating warm brine for ice melting may be made in the future.

By E. Zuckerman*

Piping Lay-Out for an Ice-Skating Rink

THE athletic association of the University of Illinois has recently completed the installation of a mechanically-refrigerated indoor ice skating rink, a notable addition to the athletic department, at a cost of $340,000.00.

The Refrigerating Equipment

The refrigerating equipment consists of the following:

2—10 in. x. 10 in. vertical enclosed twin-cylinder single-acting compressors, operating at 300 r.p.m., direct connected to 100-hp. engine-type 2300 volt synchronous motors.

4—ammonia condenser shells, size 18 in. x. 17 ft. 1½ in. arranged in two stands, two shells high, with a total condensing surface of 1080 sq. ft.

1—ammonia receiver, size 36¾ in. x. 17 ft. 7 in.

1—brine cooler, size 46¼ in. x. 15 ft. ¼ in. having a cooling surface of 2000 sq. ft.

1—suction trap, 26 in. x. 6 ft. 6 in.

1—brine storage tank, 20 ft. x 8 ft. x 8 ft.

1—brine pump, 300 gal. per min.

1—brine pump, 600 gal. per min.

The compressors are of modern design, using ring plate valves for both suction and discharge, gear oil pump for pressure lubricating system, and neat manifold incorporating stop valves, by-pass valves, and a suction strainer. Condensers are of the multipass type, using cooling water from the university plant, which, after passing through the condensers, is pumped to another portion of the campus for further use.

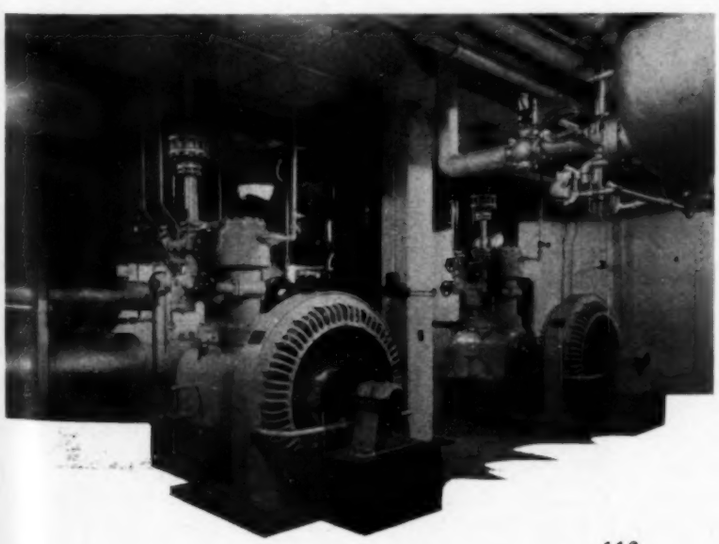


FIG. 1—THE REFRIGERATING EQUIPMENT. THE BRINE COOLER IS SUSPENDED FROM THE CEILING BECAUSE OF SPACE REQUIREMENTS

* The Carbondale Machine Co., Chicago, Ill.

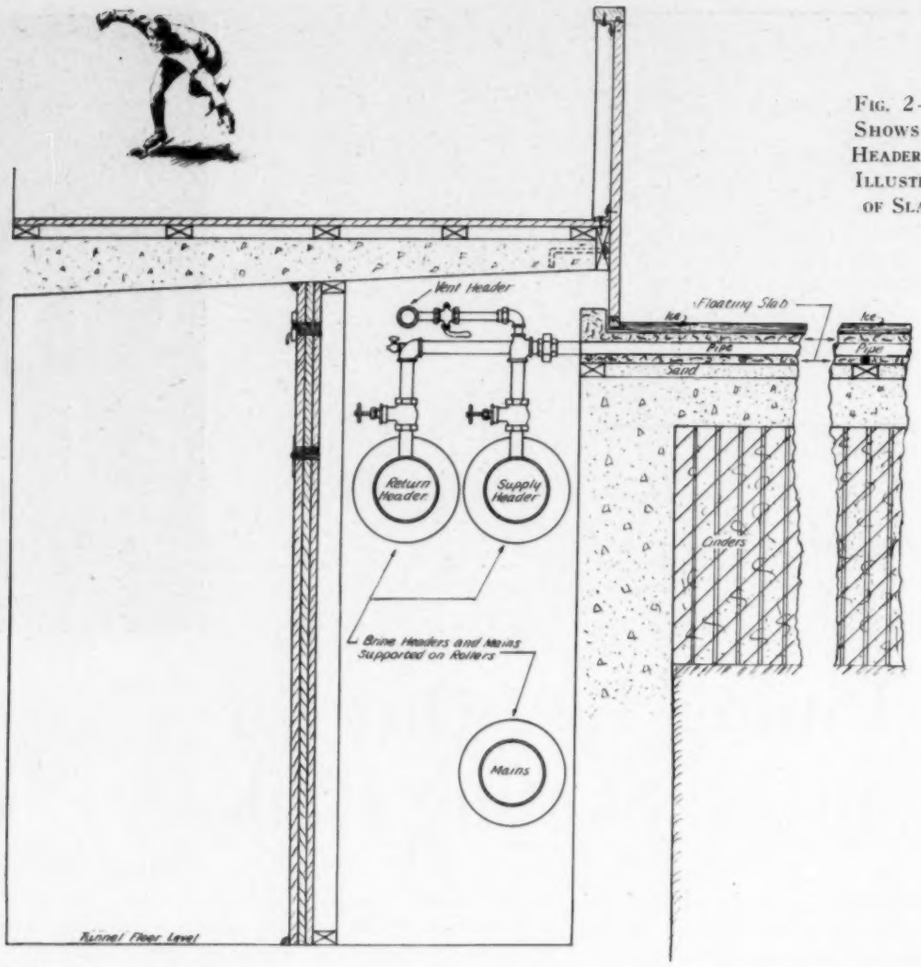


FIG. 2—THIS DRAWING SHOWS DETAILS OF THE HEADER PIPING AND ALSO ILLUSTRATES THE METHOD OF SLAB CONSTRUCTION

The brine cooler is of the multipass tubular type and is suspended from the ceiling because of space requirements. A heavily reinforced slab was provided for this purpose, strong enough to support the operating weight of the cooler, which is in the neighborhood of 16 tons. The cooler is arranged for float feed, using a low pressure float to maintain constant level irrespective of the refrigerating load or the number of compressors operated. A further precaution against slop-over is a large-size suction trap.

Either of the brine pumps, or both, may be used, depending on the refrigerating load.

Fig. 1 is a view of the machinery room.

The Piping Arrangements

The skating arena is 193 ft. long and 126 ft. wide. The cooling pipes are imbedded in the top slab. The arena has a trench along one side and the two ends. The second side adjacent to the machine room has a tunnel 6 ft. high and 6 ft. wide. Two snow pits are provided along one side of the rink for dumping scraped snow or ice.

Approximately thirteen miles of 1¼-in. pipe were used for the rink floor. The pipe was laid cross-ways of the rink on 4½-in. centers and made in hairpin bends, the tails of which have valved connections to feed and return headers in the tunnel, as shown in Fig. 2. In order to equalize friction in all hairpin bends, a three-header system was used. While this increased the length of brine travel, it makes the adjustment of individual coil valves much easier.

A valved air vent header is connected to each coil inlet, terminating in an air trap installed above the ceiling of the first floor. In addition, individual vent cocks are installed on each coil return.

The headers and connections are all galvanized as are the coil tails. No insulating material for headers and connections was used beyond a three-layer wood bulkhead, which separates the header compartment from the rest of the tunnel space. Hinged access doors are pro-

FIG. 3—THE SLAB WAS POURED IN ALTERNATE SIX-FOOT WIDE STRIPS

FIG. 4 — THE PIPING BEFORE THE SLAB WAS POURED

vided for the entire length of the bulkhead. While there is some loss of refrigeration because of uninsulated headers and connections, the probable saving was not considered sufficient to offset the first cost of the insulating material and subsequent costs at a later date, when for some reason or other access must be had to the headers and connections.

Installing the Piping

Fig. 2 also illustrates construction of the rink. The walls, trenches and tunnel were installed first. The ground was graded and filled with an 18-in. depth of cinders, on top of which was poured a 6-in. concrete sub-base, level to within ¼-in. A system of tile subdrainage was installed cross-ways of the rink.

1¾ in. x 2¾ in. creosoted screeds were laid lengthwise of the rink on 6-ft. centers, but not fastened to the sub-base. On top of these sleepers and serving as rests for the pipe were placed 1-in. solid iron rods.

Forty-foot lengths of pipe were used in order to reduce the number of field welds. All pipe lengths were welded first, this being more convenient, because the welders could roll the pipe. The bends were welded to the pipe later by raising the ends and resting them on a trestle. Approximately 2,300 welds were made by the acetylene process; only half a dozen leaked when the system was tested hydrostatically at 125 lb.

The pipes were clamped to the sleepers on exact 4½-in. centers and leveled with a transit to within 1/16-in.

Slab Is Expansion Floating Type

Clean, coarse sand was filled and tamped in to the height of the sleepers, after which the floor was ready to receive the top slab. This patented slab is known as the expansion floating type. It is entirely independent of the sub-base and the building construction, because of expansion and contraction, which, between winter and summer, varies as much as 1½-in. in the length of the rink.

The mixture for the top slab consisted of one part of cement, one and one-half parts sand, and approximately two parts of metal turnings. It was poured in approximately 6-ft. wide alternate strips cross-ways of the rink, with 1/16-in. brass expansion strips to separate the individual slabs and extending from top of slab to the top of the sand. These strips were punched so as to bond into the slab. The intermediate layers were poured later. The surface of the concrete is 1¼-in. above the top of the pipe.

The surface of each slab section was trowelled to a smooth finish. All high spots were ground smooth with a carborundum machine.

Used for Skating Only at Present

It is the intention of the athletic department, for the present, to use the arena for ice skating only. Hence, the operation will be continuous beginning the first part of November and ending about April 1. However, the arena may be used for other purposes in the future. Blank tees have been provided in the brine supply and return lines, to which a brine heater may be connected for warming the brine required to loosen the ice from the floor. The ice will then be broken up and dumped into the snow pits, where it will melt and run off into the sewer.

The average brine temperature supplied to the rink pipes is 15 F. At the start of the freeze, the temperature of the concrete slab is brought down as close as possible to the brine temperature. The slab is then sprayed with a fine film of water, which freezes rapidly. The spraying is continued until a substantial thickness of ice is reached, usually about 1½ in. When the ice thickness becomes too great due to frequent spraying, an ice planing machine is used to remove excess ice and smooth the surface. The snow formed by planing is dumped into the snow pits.

The building and equipment were designed by Holabird and Root, architects, with Professor H. J. Macintire cooperating in the design of the refrigerating plant.

Drying Method *is* All-Important

— *to dry a product economically, the proper method must be used*

The use of the proper drying method has a profound effect on the quality of the product secured and on the cost of manufacture. In order to dry a product economically, the effect of the drying factors must be known.

In this article, essential drying information is charted and discussed and the use which the industrial air conditioning engineer can make of this information is demonstrated.

By Malcolm Tomlinson*

METHOD is all important in drying processes. A vivid illustration can be had on any clear, breezy day. Double a wet blanket over a clothesline in such manner that the wind and the sun's rays strike opposite sides. During summer weather in the temperate zone the side exposed to the sun is likely to dry more slowly than the opposite side. In the torrid zone the sunny side is likely to dry first.

Although the most primitive forms of drying consisted of the application of heat alone it is amazing how many industrial drying operations are still carried on through use of this single drying factor.

In some cases where it is not practical to apply air motion or reduced air pressures, humidity control offers an opportunity for improvement in method. Many dry-

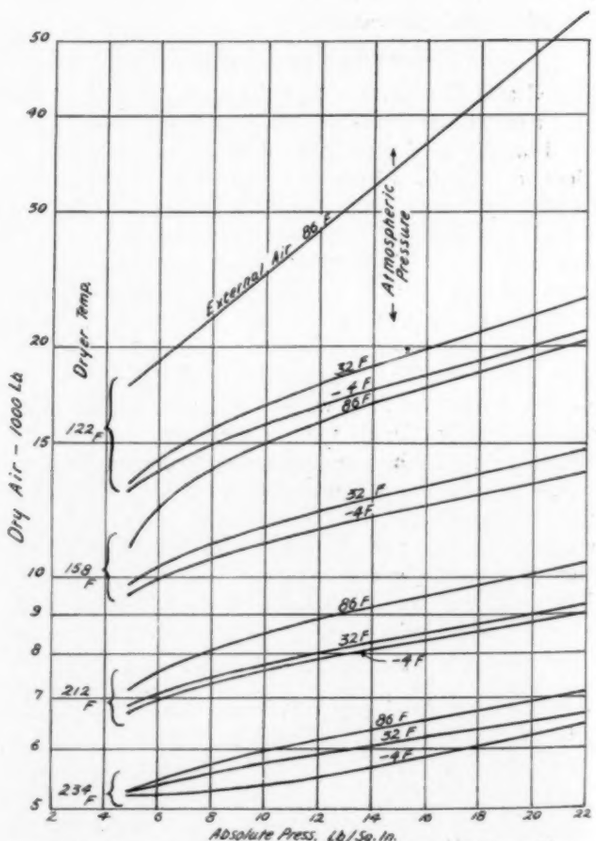


FIG. 1—WEIGHT OF AIR REQUIRED FOR DRYING UNDER VARIOUS PRESSURES

Water evaporated, 220 lb. Entering and emergent air saturated.

*Consulting Engineer, West Chester, Pa.

ing problems in the production of paper products, printing, and lithographing have been solved in this way. On the other hand humidity control offers little advantage over temperature control alone in calendering rubber sheet products. What is needed is the ability to recognize the general drying requirements of any product through a clear picture of the various drying methods and their effect on various materials.

Economical drying depends largely on the method.[1] Since practically any product can be dried in a number of different ways the relative merits of each possible method should be carefully considered before equipment is selected. In any analysis of this sort the effect of each of the four drying factors—air motion, air pressure, temperature and relative humidity—has a bearing which may not be neglected. These factors might rightly be called the mechanics of drying.

In the December, 1931, HEATING, PIPING AND AIR CONDITIONING several charts were shown. Data were plotted on the weight of moisture in air entering a dryer under atmospheric pressure with the emergent air in a non-saturated or a saturated condition, the heat required for drying under variable air pressures with the emergent air saturated and the moisture in saturated air at low temperatures under atmospheric pressure and at ordinary temperatures with variable air pressures. These graphical examples are amplified by five more charts (Figs. 1-5) published with this article in which the weight of dry air required for drying under variable air pressures and emergent air humidities as well as the heat required for similar drying conditions are shown.

Charts Show Unit Effect of Humidity, Temperature and Pressure

With these charts as a basis a new set of graphs can be made, in order to demonstrate clearly the unit effect of relative humidity, temperature and air pressure over a wide range of drying conditions in which a fixed amount of moisture is evaporated. This amount is taken as 220 pounds; the graphs appear as Figs. 6 to 11. In all of these graphs no time unit appears. The time required depends on many factors—such as method of drying, character of the product dried and rate of air motion.

Use of the Charts in Solving Drying Problems

Figs. 6 to 11 are usable in the solution of many drying problems since they show trends and comparisons. For example (by means of Figs. 6 and 7) it is possible to see what happens when moisture removal rates decrease after the initial drying period is passed. Figs. 8 and 9 give the requirements in weight of dry air and in heat per unit change in dryer temperature. Again, in Figs. 10 and 11, the heat and dry air requirements increase as the absolute pressure rises.

In these charts, the weight of dry air has been employed rather than the volume. This need not involve any hardship to those who prefer to think in terms of volume. Conversion can easily be made through the use of psychrometric charts. Most engineers prefer the weight unit in dealing with air problems because change in air temperatures means a corresponding change in volume. On the other hand a pound of dry air is not changed by temperature variations. For these reasons calculations on the basis of air volumes require more labor than those based on air weight.

[1] See "An Introduction to Drying," by Malcolm Tomlinson. HEATING, PIPING AND AIR CONDITIONING, June 1931, p. 473. Also, "Charts Show Fundamentals of the Drying Process," December 1931, p. 1017.

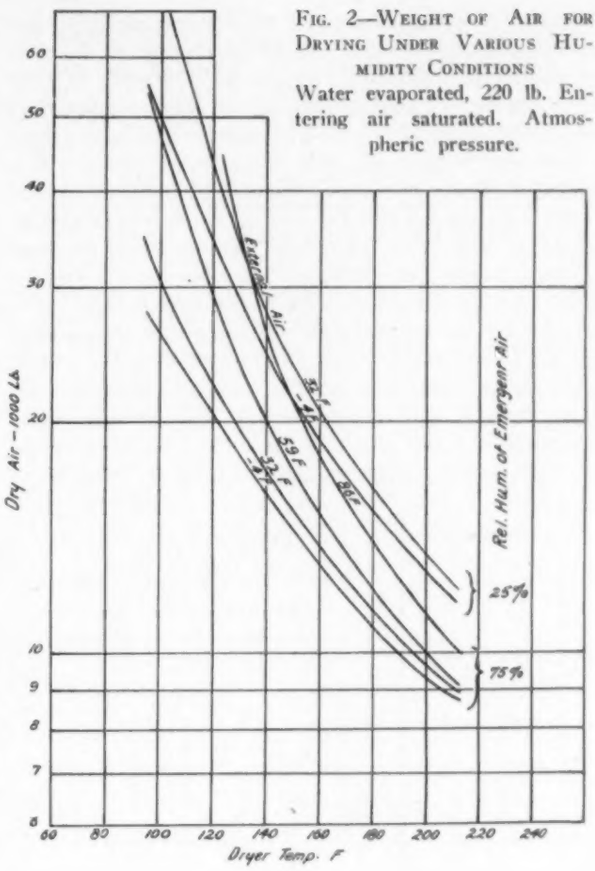


FIG. 2—WEIGHT OF AIR FOR DRYING UNDER VARIOUS HUMIDITY CONDITIONS
Water evaporated, 220 lb. Entering air saturated. Atmospheric pressure.

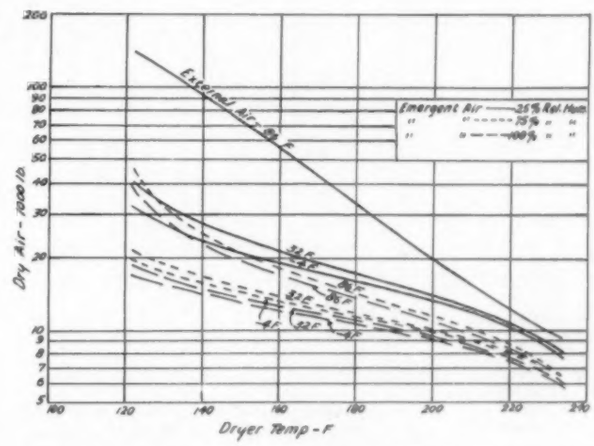


FIG. 3—WEIGHT OF AIR FOR DRYING AT VARIOUS HUMIDITIES

Drying may be considered as proceeding by one of these four methods:

1. Air drying—Pressure usually atmospheric.
2. Steam drying—Pressures seldom, if ever, atmospheric.
3. Product heated directly—Pressure that is most suitable.
4. Air conditioning—Pressures usually approximately atmospheric.

Air drying lends itself particularly to products where

slowness in drying is needed to prevent warpage and discoloration, where high temperatures can not be used and where the interior of the product is difficult to heat. Steam drying is often applied as an improvement over the previous method in *rate* of drying. Its disadvantage is that best results can only be secured where air is eliminated from the atmosphere. The third method usually employs radiation as well as conduction and convection of heat. It is used in vacuum, high pressure and high temperature drying. Its advantage lies in its speed. Air conditioning permits the variation of three of the drying factors under air pressures at or near atmospheric. Its advantage is that it can be used for a wide variety of products and for the most delicate materials. Direct heating, such as used in the third method, may be secured, in some cases, by the application of electrical currents internally while external heating, in other cases, may be obtained through exposure to the products of combustion.

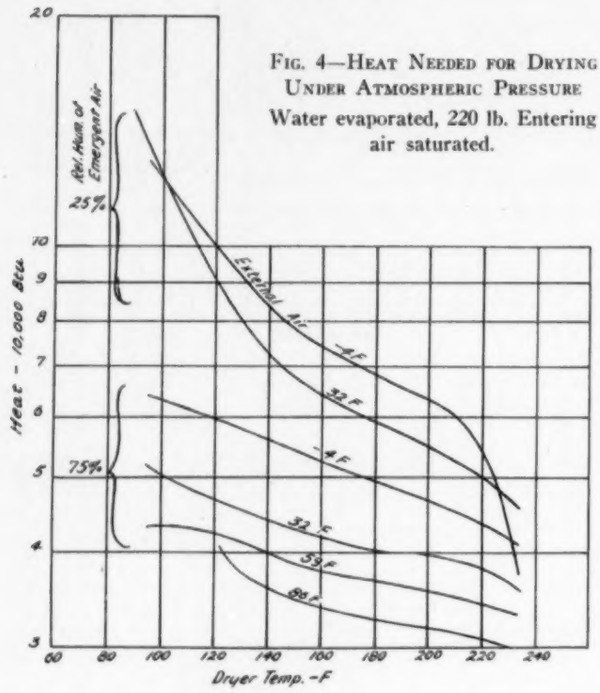


FIG. 4—HEAT NEEDED FOR DRYING UNDER ATMOSPHERIC PRESSURE
Water evaporated, 220 lb. Entering air saturated.

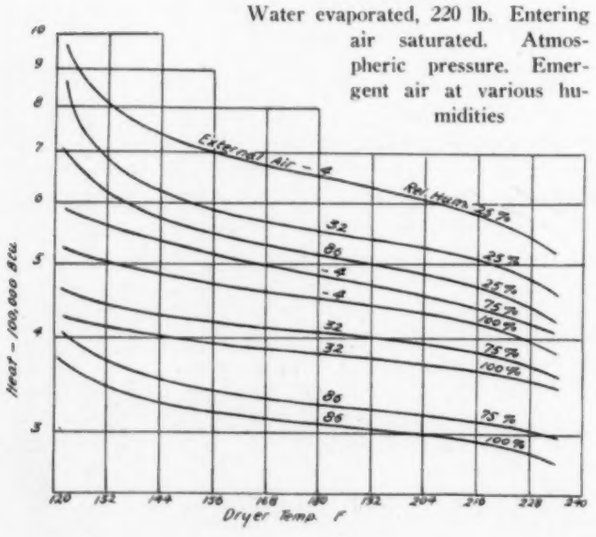


FIG. 5—HEAT FOR DRYING AT VARIOUS HUMIDITIES
Water evaporated, 220 lb. Entering air saturated. Atmospheric pressure. Emergent air at various humidities

All drying requires a preliminary knowledge of the highest permissible temperature, the weight of the product to be dried and the quantity of moisture to be removed. The next step is the determination of the quantity of heat and air needed and the most suitable air pressure. At once the charts become valuable, for comparison, under many drying combinations. From the quantity of heat required the heating surface may then be figured and the volume as well as the rate of air movement can be determined through the weight of dry air needed.

Drying with Air

Drying by the aid of air insures certain conditions. The amount of heat required is proportional to the weight of air used, since the largest expenditure of heat is to warm the air. Also the higher the air temperatures in the dryer, the lower the air requirements. The use of the optimum temperature is therefore imperative. Furthermore, the temperature of the entering and the emergent air must finally reach the same value when a bone-dry product is desired. It thus follows that where the product is not to be thoroughly dried the exit temperature will not have reached that of the entering air by the end of the drying cycle. It is, of course, imperative that the exit air retain not only the moisture which it held in vapor form when it entered the dryer but also all of the moisture which it absorbed from the product.

In addition the air at the exit has given up heat to the product. This heat must be sufficient to evaporate all of the moisture which must be removed. Again, when the temperature of the entering air is raised, the heat consumption is decreased while a diminishing external temperature raises the quantity of heat required. Then, from a weight-of-air-requirement standpoint, the air needed increases with an increase in the dryer temperature while less air is required when the outside temperature is decreased. Most of these fundamental facts will be found borne out by the charts under conditions of atmospheric pressure.

With the above facts in mind it follows that the capacity of fans to handle the air must be sufficient to take care of the greater quantity of air needed in warm weather, the heating surfaces must be sufficient to meet the coldest outside air temperatures and, where air conditioning is used, humidifying and dehumidifying equipment must be capable of meeting the extremes of humidity which occur in nature. Therefore, analysis of weather reports covering local conditions is essential.

Importance of the Proper Method

We now have a picture of the complications encountered in the selection of a satisfactory method of drying for a definite product. It is not hard to visualize the disastrous effect of poor drying methods and, on the other hand, the happy effect of right methods of drying on all types of materials and products irrespective of mineral, vegetable or animal origin.

Drying Lumber

Consider lumber, for an example. Sun, or outside air, drying is certain to result in heavy losses due to warpage,

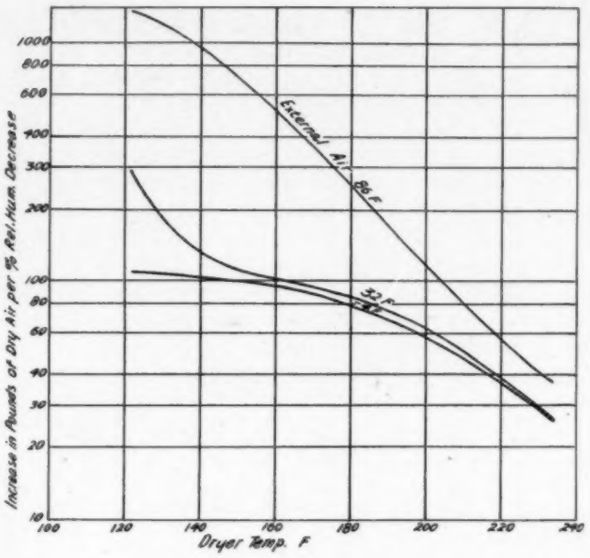


FIG. 6—EFFECT OF RELATIVE HUMIDITY ON WEIGHT OF AIR REQUIRED FOR DRYING

Entering air saturated. Atmospheric pressure. Water evaporated, 220 lb. Humidity of emergent air varying.

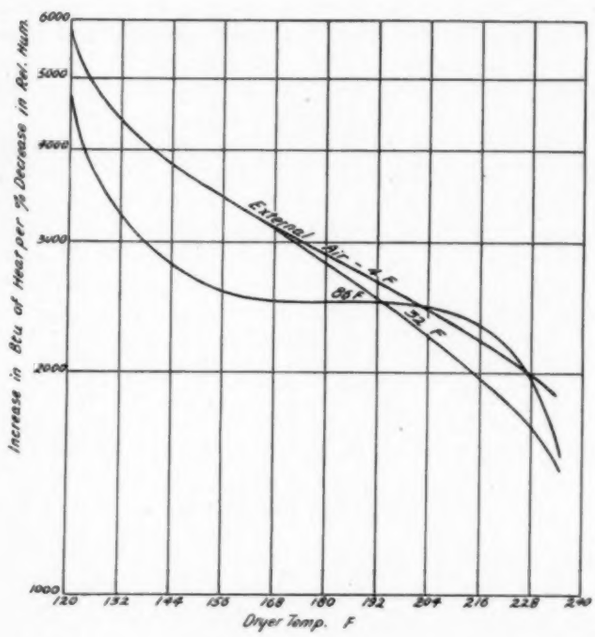


FIG. 7—EFFECT OF RELATIVE HUMIDITY ON HEAT REQUIRED FOR DRYING

Water evaporated, 220 lb. Entering air saturated. Atmospheric pressure. Humidity of emergent air varying.

cracks, shrinkage, checking, stains, fungus growth and insect attack long after drying operations are completed, because only a small portion of the moisture, or "sap," has been removed. Kiln drying with temperature control alone means that the rate of evaporation may not be controlled properly and the surfaces dry faster than the interior. This condition is known as "case hardening" and results in surface checking. To overcome this condition steam drying or air conditioning is used. In other words the air surrounding the lumber must be kept

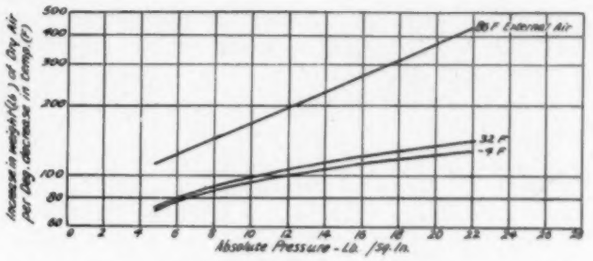


FIG. 8—EFFECT OF TEMPERATURE (DRYER) ON WEIGHT OF AIR FOR DRYING

Water evaporated, 220 lb. Entering and emergent air saturated.

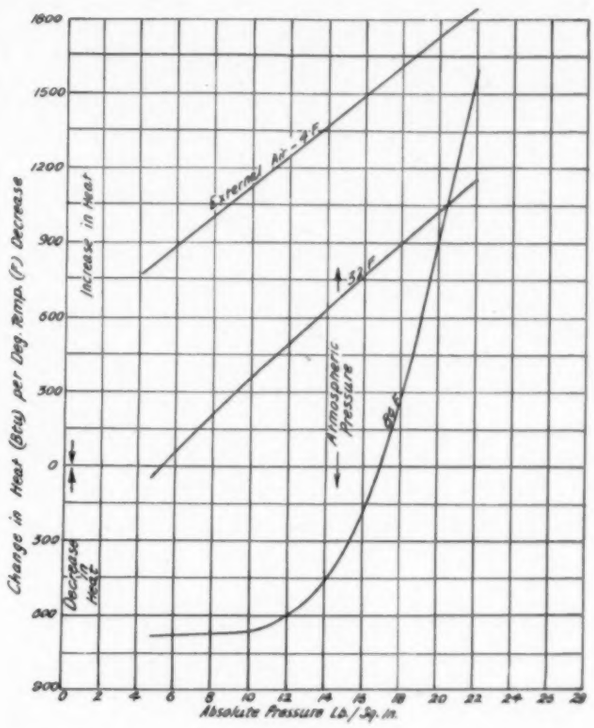


FIG. 9—EFFECT OF TEMPERATURE (DRYER) ON HEAT FOR DRYING

Water evaporated, 220 lb. Entering and emergent air saturated.

humid, or wet steam must fill the atmosphere to enable interior moisture to leave the boards before the surface is sealed. Kiln production of hard and soft woods is about 22 billion board-feet per year. Losses due to improper outdoor and subsequent kiln drying are estimated at 7 per cent of this production. Proper kiln drying in the green condition would reduce the losses to not more than 2 per cent, it is estimated. The difference would represent an annual saving of at least $25,000,000 to the lumber business. To the customer it would mean a reduction of from 2 to as much as 30 per cent in freight charges, a superior product on which wastage would be markedly reduced, and, therefore, a lower price level for the ultimate product. Many manufacturers of high-grade wood products have their own kiln drying plants in order to insure a supply of well-dried wood.

The example cited covers a material in which diffusion

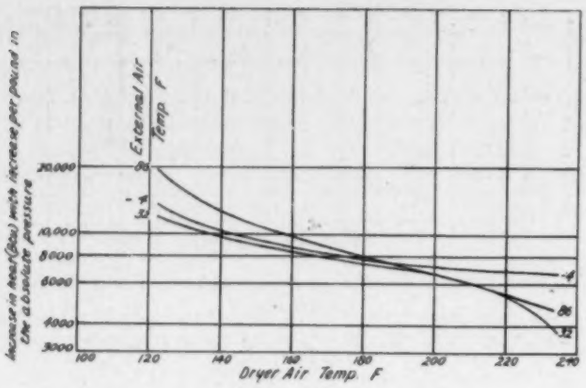


FIG. 10—EFFECT OF PRESSURE ON HEAT FOR DRYING
Water evaporated, 220 lb. Entering and emergent air saturated.

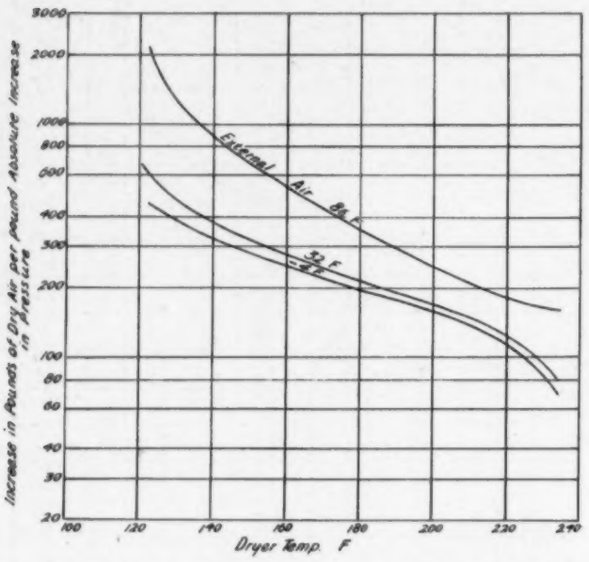


FIG. 11—EFFECT OF PRESSURE ON WEIGHT OF AIR FOR DRYING
Water evaporated, 220 lb. Entering and emergent air satrated.

from the interior is not as rapid as surface evaporation and where shrinkage and skin effect are important. With thin sheets of paper diffusion and surface evaporation are practically merged. Shrinkage and color fading become important items. The moisture content of such paper products, when exposed in single sheets to the atmosphere, changes instantly with the slightest changes in humidity. Here heat and humidity must be carefully controlled. Where the thin sheets are in rolls the paper may be dried quickly with a low vacuum which makes possible the use of temperatures about 100 degrees higher than can be used under atmospheric pressure.

Physical Structure of the Product Influences the Method

The shape and physical structure of each product thus markedly influences moisture diffusion from its interior, surface evaporation, shrinkage and skin effect. Indirectly these four physical factors determine the drying rate and therefore the choice of the combination of drying factors which will give the best drying methods.

Shanghai Has Air Conditioned Building

By Cyril L. Tilburn*

THE Sassoon House, Shanghai, China, is a 12-story building, the largest in the Far East, covering the end of a block about 50,000 sq. ft. in area and situated in the hub of the business and shopping districts. It stands as a feature of the rapid development of western civilization in a land famed for its adherence to tradition. From its technical aspects this structure has been a distinct step towards undertakings on an unprecedented scale following western ideas.

The building houses banks and shops on the ground and mezzanine floors with additional interior frontage to three arcades which pass through the block. The first to third floors contain offices of numerous business firms, in addition to those engaged in professional work, and the Chinese government radio station operates from here also. The remaining portion of the building is arranged as a 250-room hotel.

Forced Hot Water Heating

The heating system is an accelerated hot water, two-pipe one. All circuits are divided into separate mains for controlling banks, shops, offices, and hotel separately on account of the variety of these tenancies. The system of heating is arranged on two levels with ground to third floor separate from the remainder of the building. Although the principles are the same the method of valving circuits are appreciably different.

On account of the mobility that rentable spaces demand for the alterations that usually occur, with the partitioning arrangement varying with the type of tenant, valves were fixed on all risers so that alterations are made without inconvenience to service. In the case of the hotel the mains were divided into eight sub-circuits controlling equal areas. All piping serving the low level passes through the "raft" which extends under the whole area of the building, with mains arranged in suitable vertical pipe shafts in the usual way.

Domestic Hot Water

Three 1250-gal. storage tanks are arranged for hot water supply to the hotel, and one 1000-gal. tank for the balance of the building. Heat generation is by the usual battery of steam coils with temperature ranges thermostatically controlled. All condensate drainage passes through steam traps to a common condensate tank and is returned by pumps to the steam boilers.

The Boiler Plant

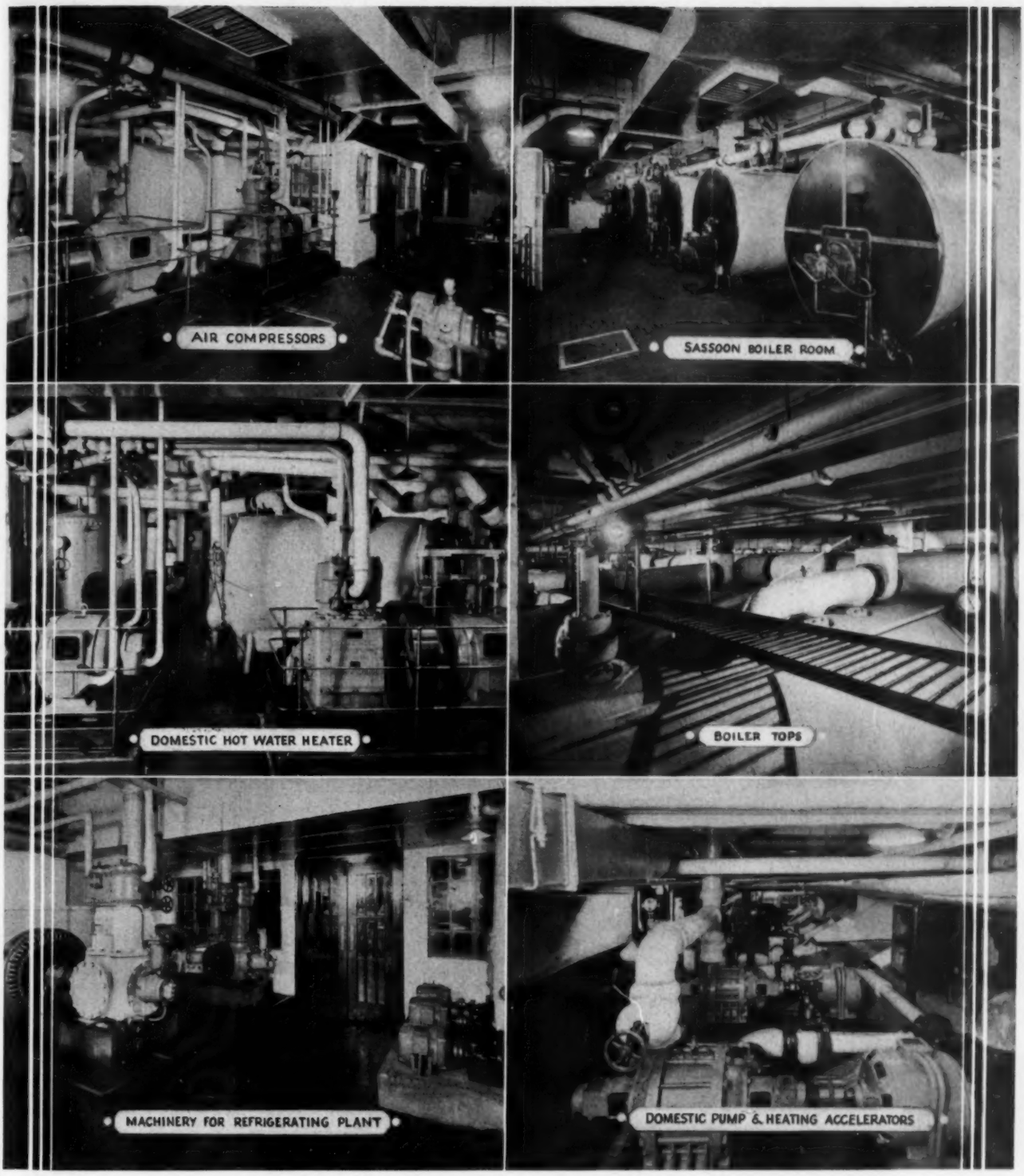
Six 3,000,000-Btu boilers are arranged as:

(*A*) 2 Steam
(*B*) 2 Steam, or direct hot water heating
(*C*) 2 Hot water heating

Their uses are:

(*A*) Normally two boilers are in operation, serving steam to the kitchens and hotel services, in addition to the usual domestic requirements. During winter months a third boiler is in service to handle

*Engineer-Superintendent, Sassoon House, Shanghai.



the extra demands for heating.

(*B*) Spares to *A* and *C*.

(*C*) Heating from ground to third floors.

All boilers are steel flue and return tube type of 3,000,000 Btu capacity each, and are oil-fired.

Refrigeration Plant

A central refrigeration system is arranged with cold rooms and boxes literally scattered throughout the building, with branches to tenants outside of the hotel. The system is brine-cooled. Refrigeration is by means of ammonia compressors in the basement adjacent to the boiler room. All other mechanical equipment with the exception of pumps, such as condensers, brine coolers, etc., is arranged in the backyard a few feet away.

The refrigerating machinery consists of three units inter-connected to serve the central refrigerating system or the air cooling system of the air conditioning plant.

Air Conditioning Plant

Air washers are arranged in two sections of the eighth floor which supply air conditioning to public rooms of the hotel situated on the eighth, ninth and tenth floors. All equipment is hand-controlled.

Loss in Elbows and Bends

▼

Heat Sensitive Paint

"Open for Discussion"

IN THE article "Piping Improvements Reduce Cost of Pumping," (HEATING, PIPING AND AIR CONDITIONING, October, 1931, page 836), the velocity head is calculated and assigned to a length of 100 ft. This method is in error because the velocity head is present whenever water is flowing with a certain velocity and is entirely independent of the length of pipe.

Then I notice also that loss in head due to an elbow is calculated as $0.46\ v^2/2g$.

I think this is too low for the elbow shown in the illustrations. As a general rule the loss of head in a cast iron elbow is $v^2/2g$.

There is a difference between a pipe bend and an elbow. In a pipe bend the loss of head is produced by the change in the direction of flow and by the friction within the pipe. In an elbow the loss of head is produced by the change in the direction of flow, by the friction within the pipe, and by the friction resulting from the sudden enlargement of the path from the pipe to the elbow and the sudden contraction from the elbow to the pipe.

The loss of head in a pipe bend varies with the radius of the bend. The values in "Hydraulics" (King and Wisler) quoted in the article, are of the losses of head in pipe bends produced by the change in the direction of flow; to these values must be added the loss of head produced by the friction within the pipe, in order to secure the total loss of head in the pipe bend. For example, for a pipe bend having a radius of 2 ft., the book gives a value of $0.28\ v^2/2g$; this is equal to 0.44 ft. for a velocity of 10 ft. per sec. The length of the pipe bend is about 3.14 ft.; for this length and a velocity of 10 ft. per sec. and a 1-in. pipe, the friction head is about 1.31 ft. The total loss of head in the pipe bend is therefore 0.44 plus 1.31, or 1.75 ft.

It should be evident from these explanations that the values of coefficients quoted in King and Wisler's "Hydraulics" for pipe bends cannot be used to represent the losses of head in ordinary elbows, such as are shown in Fig. 2 of the article, page 838, and referred to in the corresponding calculations of the article in question. If we place the loss of head in these elbows equal to $0.46\ v^2/2g$ we make an error of at least 50 per cent.

Readers interested in losses of head in elbows can secure valuable information from Bulletin 222, Engineering Experiment Station, University of Illinois. —*F. E. Giesecke.*

REFERRING to the article on a heat sensitive paint which appeared on page 86 of the January HEATING, PIPING AND AIR CONDITIONING, I claim no credit for the paint described. I was badly in need of such a paint when developing the steam cooling system for automobiles, as I had found it impossible to determine how the steam was distributed throughout the radiator, or condenser. I tried for several years to find some chemical that would change color with temperature without success until I saw an article by W. S. Andrews.

It seems to me this paint should be very useful in every laboratory or power plant, to indicate unmistakably whether a surface is hot or cold.—*S. W. Rushmore.*

Conventions and Expositions

American Oil Burner Association: Annual convention, April 11-16; Mechanics Building, Boston, Mass. Secretary, H. F. Tapp, 342 Madison ave., New York City.

Heating and Piping Contractors National Association: Annual convention, May 16-18; Book Cadillac Hotel, Detroit, Mich. Secretary, Joseph C. Fitts, 50 Union Square, East, New York City.

National Electric Light Association: Annual convention, June 6-10; Atlantic City Auditorium, Atlantic City, N. J. Secretary, A. J. Marshall, 420 Lexington ave., New York City.

National District Heating Association: Annual convention, June 14-17; William Penn Hotel, Pittsburgh, Pa. Secretary, D. L. Gaskill, Greenville, Ohio.

American Society for Testing Materials: Annual meeting, June 20-24; Atlantic City, N. J. Secretary, C. L. Warwick, 1315 Spruce st., Philadelphia, Pa.

American Society of Mechanical Engineers: Spring meeting, June 27-July 1; Bigwin Inn, Lake-of-Bays, Ontario, Canada. Secretary, Calvin W. Rice, 33 W. 39th st., New York City.

Heating · Piping
and Air Conditioning

American Society *of* Heating *and* Ventilating Engineers

JOURNAL SECTION

FEBRUARY *1932*

Material in this Section prepared by Publication Committee: W. A. Rowe, *Chairman;* E. O. Eastwood, and F. D. Mensing A. V. Hutchinson, Manager of Publications

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Field Studies of Office Building Cooling

By J. H. Walker[1], S. S. Sanford[2] (*MEMBERS*) and E. P. Wells[3] (*NON-MEMBER*)
Detroit, Mich.

Report of research conducted in co-operation with the Committee on Research of the American Society of Heating and Ventilating Engineers

THERE is a tendency in ventilating system design to copy previous work without an adequate check-up of actual performance. A great many mistakes could doubtless be avoided, particularly in a relatively new field such as building cooling, if design methods were carefully correlated with field experience.

The tests described in this paper were made because the authors felt that some actual results of the performance of a building cooling system would be of considerable value to designers of such systems. Because the tests were conducted under conditions which approximated ordinary operating conditions, it was impossible to control some of the factors as accurately as would be done in a laboratory test, and this should be borne in mind in making use of the results.

The work was done in the Union Guardian Building in Detroit, shown in Fig. 1. It is a 40-story building of which the lower 16 floors and two basements are artificially cooled. The air-conditioning system has been previously discussed,[4] but the following is a brief description of the important features. The building is of more or less typical office building construction having brick walls, backed up with hollow tile, and with two large steel-sash windows per bay. The heating as well as the cooling requirements are taken care of by a fan system. Air enters each room through one or more ceiling diffusers which discharge horizontally in all directions except towards the windows and the exhaust is taken care of by grilles in the office partitions through which the air finds its way to the corridor and thence to the two recirculating risers. The system is zoned and each zone is equipped with a variable speed fan, making it possible to control the temperature on different sides of the building in summer by varying the volume of air supplied. Data regarding the building and air conditioning system are given in Table 1.

The maximum weather conditions in Detroit for which cooling systems are designed are approximately 95 F dry bulb and 75 F wet bulb. The Union Guardian Building is used for banking, trust company and tenant office purposes. The temperature carried on the office floors varies from 73 F dry bulb and 60 F wet bulb to 79 F dry bulb and 64 F wet bulb, depending upon the outside temperature. The temperature in the lobby is about one degree higher than in the offices and that in the basements about two degrees lower than in the offices.

The various factors which comprise the cooling and dehumidifying load of a building are as follows:

I. Load due to weather conditions:
   1. Heat transmission through walls and roof.
   2. Heat transmission through window glass.
   3. Sun radiation through window glass.
   4. Heat and moisture in make-up and leakage air.

FIG. 1—UNION GUARDIAN BUILDING

[1] Superintendent of Central Heating, The Detroit Edison Company.
[2] Sales Engineer, The Detroit Edison Company.
[3] Air Conditioning Engineer, Union Guardian Building.
[4] Air Conditioning System of a Detroit Office Building, by H. L. Walton and L. L. Smith (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929); also, Operation of an Air Conditioning Plant in a Large Office Building, by Earl P. Wells, (*Heating & Ventilating*, September 1931, p. 62).

Presented at the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1932.

II. Load due to occupancy:
   5. Heat from lights.
   6. Body heat and moisture from occupants.
   7. Heat and moisture from machinery, piping and processes.

The tests herein described were confined to one factor in the total cooling load of the building, namely, that due to external weather conditions, and involve items 1, 2 and 3, which are the items of perhaps the greatest uncertainty in cooling calculations. The studies comprised several phases as follows:

A. Determination of cooling requirements of the four exposures of the building.

B. Measurement of the flow of heat through the building wall.

C. Studies of the temperatures of the wall.

D. Studies of the overall cooling load of the building.

Determination of the Cooling Requirements of the Four Exposures of the Building

The cooling requirements of the four sides of the building were determined by taking four rooms of identical size and exposure on the different sides and measuring the temperature and quantity of cooled air which was required to maintain the rooms at an equal and constant temperature throughout the working day. These rooms were specially constructed test rooms on the twelfth floor of the building and were located as shown in Fig. 2.

TABLE 1—UNION GUARDIAN BUILDING—PHYSICAL DATA

Building height	40 stories
Ground area	80 ft x 270 ft
Air-conditioned floors	16 lower floors and 2 basements
Air-conditioned volume	2,962,600 cu ft including toilets, inside stairs, and elevator shafts
Air-conditioned area	225,986 sq ft rentable area
Number of occupants in air-conditioned area	1850
Refrigeration capacity	600 tons
Compressors	3—200 ton CO_2
Compressor motors	3—300 hp 4600 v direct drive, synchronous
Dehumidifiers	5 with direct expansion evaporating coils
Pumps	5 spray and 5 flood
Fans	10 circulating (no general exhaust fans)
Condensers	Double-pipe, using city water

They were approximately 14 ft 8 in. wide and 17 ft deep, taking in almost the width of a bay of two windows, and had walls made of ½ in. rigid insulation nailed to wooden studding. The exterior of one of the rooms is shown in Fig. 3. There were two openings in each room, one being a small doorway through which observers entered the room and which was kept closed except when someone was passing through it. The other opening, located about 18 in. from the ceiling and to one side of the door, was the air outlet from the room and was approximately 21 in. x 12 in.

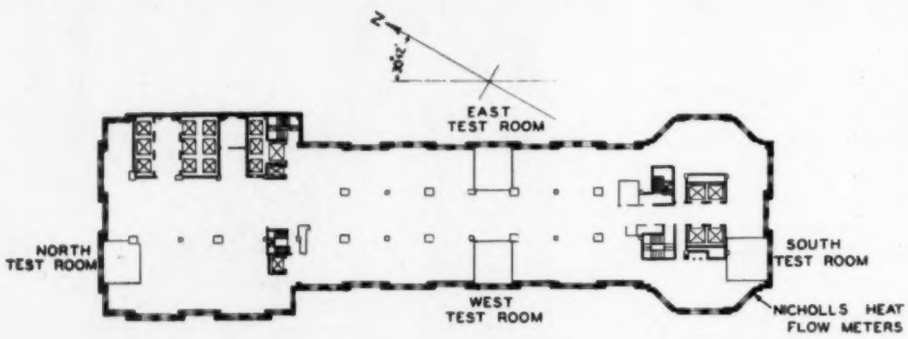


FIG. 2 — FLOOR PLAN OF BUILDING SHOWING TEST ROOMS

FIG. 3 — EXTERIOR OF TEST ROOM

FIG. 4—INTERIOR OF TEST ROOM

The interior of one of the test rooms is shown in Fig. 4, and a section and the elevation of the outside wall in a test room is shown in Fig. 5. The windows are equipped with Venetian blinds which were adjusted during the tests so as to cover the upper half of the window, with the slats in a horizontal position. The windows are of plate glass ¼ in. thick, and the window frames and sash are of steel.

Normally, air is delivered to the offices through square ceiling diffusers, two to each bay. In the test rooms, the ceiling diffusers were removed because it was difficult to regulate and measure the air coming from them and one of the openings in the ceiling was plugged. An 8 in. circular sheet-iron duct with a butterfly damper in it was attached to the other ceiling opening and all of the air was delivered to the room through this. The duct extended to the floor and back toward the ceiling as shown in Fig. 4, and the volume of air was determined by an anemometer held in the outlet of the duct.

The floor and ceiling construction in the building is as shown in Fig. 6. In the space between beams are the air ducts. The temperature in the furred space above the test rooms was read with thermometers thrust up through holes drilled in the ceiling; that in the furred space beneath the test rooms was obtained by thermometers through the ceiling below. The temperature outside the building wall was obtained from a thermocouple shielded by an insulated container having a bright metal surface with the container arranged for a free flow of air up through it.

The temperature of the entire twelfth floor was maintained as nearly as possible equal to that in the test rooms. The temperature above the ceiling ranged from 72 to 76 F and that below the floor of the test rooms ranged from

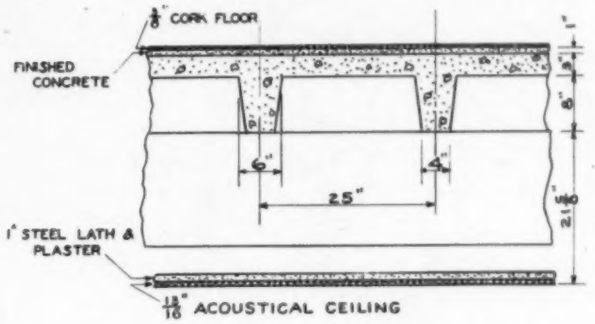


FIG. 6—CROSS-SECTION OF FLOOR AND CEILING

74 to 76 F. Except for the error caused by the difference in temperature between the test rooms and the space above and below, the heat removed from the room, as indicated by the quantity and temperature rise of the air supplied, should be substantially equal to the heat entering the room through the exposed side. In calculating the net amount of heat entering the test rooms, a correction was made for differences in temperature

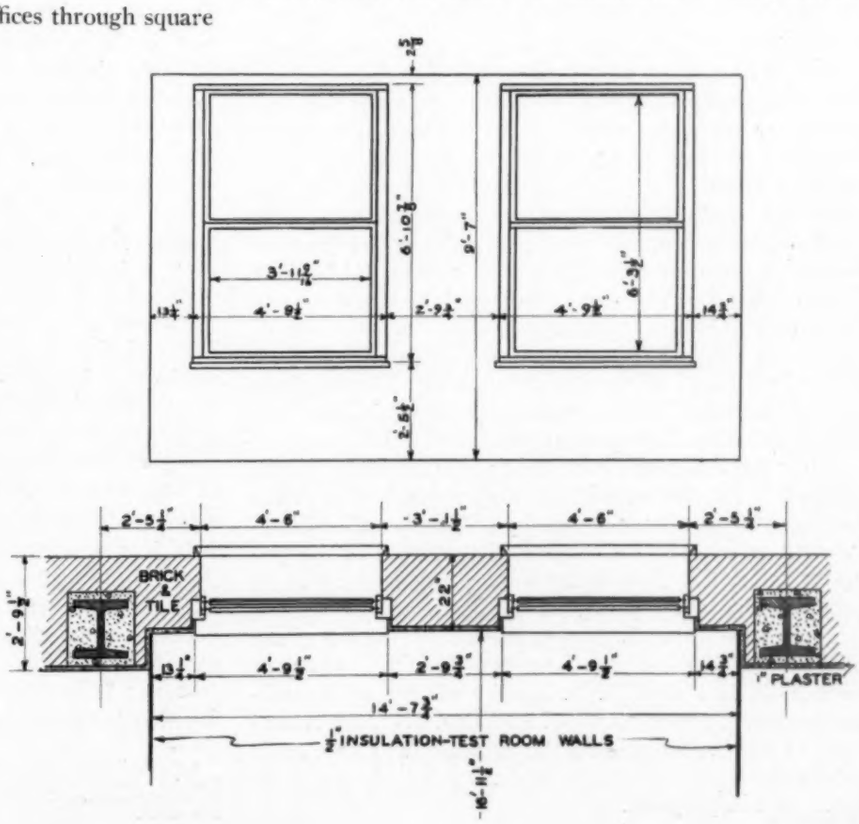
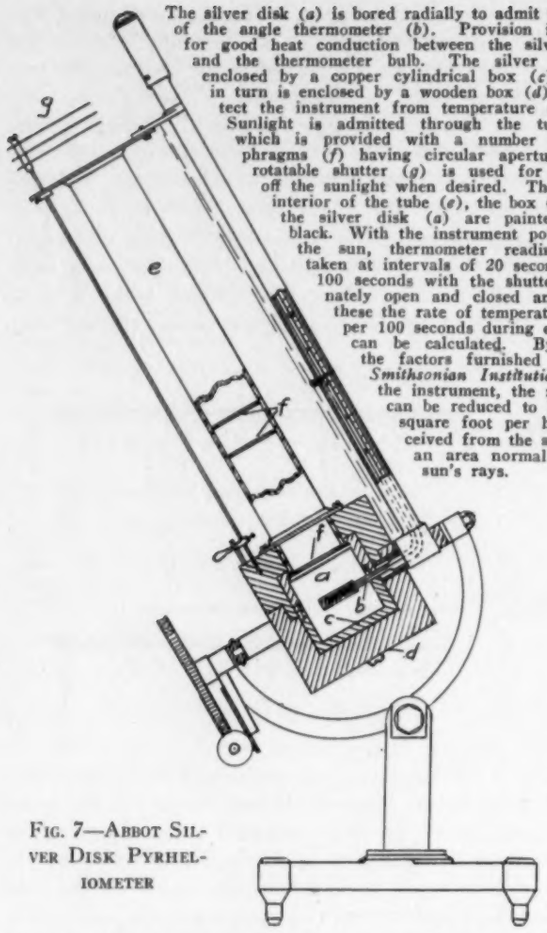


FIG. 5—ELEVATION AND SECTION OF WALL, WEST TEST ROOM



The silver disk (*a*) is bored radially to admit the bulb of the angle thermometer (*b*). Provision is made for good heat conduction between the silver disk and the thermometer bulb. The silver disk is enclosed by a copper cylindrical box (*c*) which in turn is enclosed by a wooden box (*d*) to protect the instrument from temperature changes. Sunlight is admitted through the tube (*e*) which is provided with a number of diaphragms (*f*) having circular apertures. A rotatable shutter (*g*) is used for cutting off the sunlight when desired. The entire interior of the tube (*e*), the box (*c*), and the silver disk (*a*) are painted dead black. With the instrument pointed at the sun, thermometer readings are taken at intervals of 20 seconds and 100 seconds with the shutter alternately open and closed and from these the rate of temperature rise per 100 seconds during exposure can be calculated. By using the factors furnished by the *Smithsonian Institution* with the instrument, the readings can be reduced to Btu per square foot per hour received from the sun over an area normal to the sun's rays.

FIG. 7—ABBOT SILVER DISK PYRHELIOMETER

between the test rooms and the surrounding space in the building. Obviously, no extreme degree of accuracy can be claimed for these tests. The air temperature through the room varied somewhat as did the surrounding temperatures and there were other small sources of error; but, bearing in mind that they are field tests and not laboratory tests, the method used should be acceptable.

It was, of course, anticipated that a large portion of the load would be due to sunshine through the window glass and provision was therefore made to measure the solar radiation, with the idea of arriving at some correlation between solar radiation and cooling load. The instrument used was the Abbot pyrheliometer, shown in section in Fig. 7. It operates on the principle of measuring the rise in temperature of a block of silver which is exposed to the sun's radiation. The readings can be expressed in Btu per square foot per hour, the area being taken in a plane normal to the sun's rays.

After the tests were completed the anemometers were calibrated for the conditions under which they were used by passing a known quantity of air through a circular duct which was a duplicate of that used in the test rooms and noting the relation between anemometer readings and the true quantity of air flowing.

Pyrheliometer readings were taken from the top of the south tower of the Union Guardian Building. On September 10, 11 and 12, 1931, a series of days with maximum outdoor temperatures of 90 F and over, determinations of solar radiation with the pyrheliometer were made frequently throughout each day during the time the room tests were in progress.

In order to determine the solar radiation entering the rooms through the window glass, observations from within a room were made with the pyrheliometer with the window alternately open and closed. These readings indicated that from 12 to 20 per cent was cut off by the glass. Experiments by F. C. Houghten and Carl Gutberlet[5] indicated an absorption by glass of from 8.9 to 16.5 per cent and experiments by the U. S. Bureau of Standards referred to in the same paper showed the absorption of window glass to be 18 per cent. Of the radiation which does not pass directly through the glass, part is absorbed and part reflected. Of that which is absorbed, part is delivered by the glass to the room. It was therefore assumed that of the total solar radiation, 93 per cent was delivered to the room and the other 7 per cent reflected or transferred by the glass to the outdoor air.

Results of Tests

Pyrheliometer readings for Friday, Sept. 11, 1931, are shown in Fig. 8. The time shown in this figure and throughout the paper is Eastern Standard (75th Meridian) time.

The results of the cooling tests in the four test rooms for Sept. 11, 1931, are given in Tables 2 and 3 and in Fig. 9. Similar tests were made on other days but the data for Sept. 11 are especially important because this was a day of continuous bright sunshine. In Fig. 9 the data are plotted separately for each test room. The *actual cooling load* is the rate of removal of heat from the room by the cooling air as determined by the anemometer readings and the rise in temperature of the air in passing through the room. As defined here it did not include the dehumidifying load due to moisture from people and from outdoor air. The volume and temperature of the air are given in Table 3. *Conduction, glass* is the calculated rate of heat transmission through the glass in the two test room windows based upon Weather Bureau temperatures and observed room temperatures 5 ft from the floor. The U. S. Weather Bureau station is located two blocks from the Union Guardian Building. *Sunshine through glass* is the calculated rate at which solar radiation enters the test rooms through the two windows. It was obtained by reducing the solar radiation as observed out-of-doors by 7 per cent and multiplying this by the area of sunshine on a plane normal to the sun's rays. The *net calculated load* is the algebraic sum of the heat entering the room by conduction through the glass, solar radiation through the glass, transmission through the outer wall, heat from the observers during the few minutes they were in the test room and heat entering or leaving the test room because of difference in temperature between the room and the surrounding space in the building.

It is noteworthy that the amount of heat entering through the wall was too small to plot in Fig. 9. The Nicholls' heat meter readings for this day on a southerly-exposed wall indicated that the flow of heat from the wall to the room was only about 1.7 Btu per square foot per hour when the room temperature was 75 F. On this basis the heat from the wall to the room was

[5] Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A. S. H. V. E. TRANSACTIONS, Vol 36, 1930).

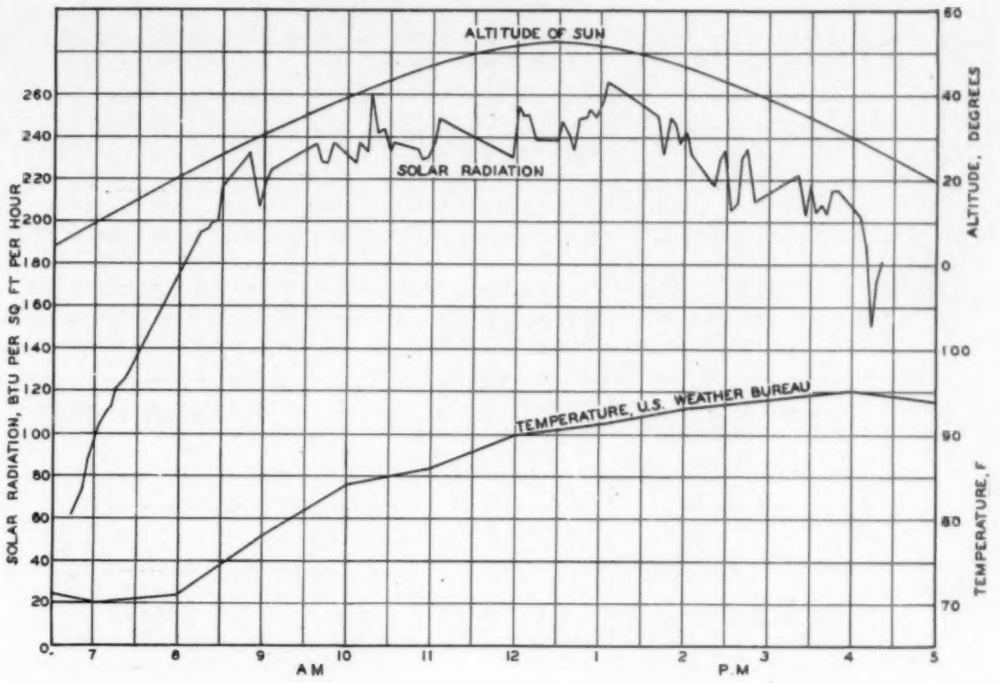


FIG. 8—SOLAR RADIATION AND OUTDOOR TEMPERATURE ON A CLEAR DAY, SEPT. 11, 1931

calculated to be 125 Btu per hour in the east and west rooms and 135 Btu per hour in the north and south rooms, these rooms having a slightly larger wall area. The heat from the wall was assumed constant although it did appear to decrease somewhat during the day.

No allowance was made for infiltration. On the day of the test the wind velocity did not exceed 12 mph and it was assumed that the pressure in the test rooms was sufficient to prevent infiltration. Air which may have leaked outward was assumed to have the same temperature as that leaving the room through the outlet and was included on that basis in the computation of the cooling load.

Discussion of Results

The rapid fluctuation in intensity of solar radiation shown in Fig. 8 may be attributed partly to changes in atmospheric conditions and partly to errors in observation. The seasonal variations in solar radiation do not appear to be great if days having a clear bright sky are compared. Readings taken during the middle of the day over the period from August 15 to November

TABLE 2—ATMOSPHERIC CONDITIONS, SEPTEMBER 11, 1931

Eastern Standard Time	U. S. Weather Bureau					Solar Radiation (Btu per Sq Ft per Hour)	Indoor Conditions, 12th Floor		
	Temperature (F)	Wet Bulb (F)	Relative Humidity (Per Cent)	Wind Velocity (mph)	Wind Direction		Dry Bulb (F)	Wet Bulb (F)	Relative Humidity (Per Cent)
1 a. m.	78	.....	.....	9	S W	0	.....	.....	.....
2	78	.....	.....	10	S W	0	.....	.....	.....
3	78	.....	.....	9	W	0	.....	.....	.....
4	76	.....	.....	9	W	0	.....	.....	.....
5	74	.....	.....	8	W	0	.....	.....	.....
6	72	.....	.....	8	S W	0	.....	.....	.....
7	70	.....	.....	9	S W	95	.....	.....	.....
8	71	67.4	83	11	S W	173	.....	.....	.....
9	78	.....	.....	8	S W	220	74.5	62.5	51
10	84	.....	.....	7	S W	235	74.5	63.0	53
11	86	.....	.....	7	S W	235	74.0	62.0	51
12	90	74.1	48	9	S W	245	74.5	61.5	48
1 p. m.	91	.....	.....	11	S W	255	75.0	62.0	48
2	93	.....	.....	12	S W	240	75.0	62.0	48
3	94	.....	.....	9	S W	215	75.0	62.0	48
4	95	.....	.....	11	S W	197	74.0	62.0	50
5	94	.....	.....	10	S W	.....	.....	.....	.....
6	93	.....	.....	9	S W	.....	.....	.....	.....
7	90	.....	.....	7	S W	0	.....	.....	.....
8	87	73.0	50	9	S W	0	.....	.....	.....
9	84	.....	.....	9	S	0	.....	.....	.....
10	80	.....	.....	8	S	0	.....	.....	.....
11	80	.....	.....	8	S W	0	.....	.....	.....
12	80	.....	.....	6	S W	0	.....	.....	.....

TABLE 3—DATA FROM TEST ROOMS, SEPTEMBER 11, 1931

Test Room	Time	Dry-Bulb Temperature, Deg Fahr: Outdoors near Wall	5 Ft from Floor	1 Ft from Floor	2 Ft from Ceiling	Average Outside the Room	Furred Space in Ceiling	Furred Space Beneath Floor	Air Entering	Air Leaving	Volume of Air Entering Room, cfm	Heat Removed from Room by Air, Btu per Hr	Sunshine Through Window Glass: Normal Area, Sq Ft	Btu per Sq Ft per Hr	Btu per Hr	Heat Through Glass by Conduction, Btu per Hr	Heat from Surrounding Space, Btu per Hr	Net Heat Entering Room, Btu per Hr[c]
East	7 A. M.	.....	76.0	75.5	75.3	76.5	74.0	.....	66.9	75.8	.....	.....	30.5	88	2685	-460	-84	2368
	8	.....	75.0	74.5	74.5	75.8	73.5	74.6	64.9	74.5	231	2435	22.0	161	3540	-290	162	3639
	9	.....	75.0	74.8	74.9	75.4	73.0	.....	65.0	74.8	204	2198	12.5	205	2560	210	-68	2929
	10	.....	75.0	74.4	74.5	75.0	73.0	74.6	65.6	74.7	175	1748	2.8	219	615	670	-103	1409
	11	.....	74.9	74.2	74.5	75.0	73.0	.....	64.5	74.8	88	996	0	0	0	850	-29	1048
	12	.....	75.1	75.0	74.9	75.3	73.0	74.8	65.0	75.1	75	832	0	0	0	1110	-161	1176
	1 P. M.	.....	75.0	74.5	74.4	75.3	73.0	.....	64.9	75.0	87	945	0	0	0	1190	-42	1375
	2	.....	75.0	74.8	74.8	75.5	73.0	74.3	65.0	75.0	93	1020	0	0	0	1350	-135	1442
	3	.....	74.9	74.2	74.5	75.5	73.0	.....	65.0	74.7	128	1368	0	0	0	1440	85	1752
	4	.....	75.0	74.8	74.8	75.5	73.0	74.5	64.9	74.9	74	812	0	0	0	1500	-47	1685
	5	.....	74.8	74.3	73.8	75.0	73.0	.....	64.9	74.8	112	1218	0	0	0	1430	21	1678
South	7 A. M.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	16.0	88	1410	-480	.....	.....
	8	.....	76.5	77.0	76.5	75.3	76.0	76.2	68.0	75.5	229	1885	22.0	161	3540	-430	-544	2800
	9	83[a]	77.0	77.2	77.2	75.4	75.5	.....	66.5	75.5	320	3160	26.6	205	5455	80	-921	4845
	10	87.5	77.4	77.5	77.5	75.7	75.0	75.0	66.0	75.9	348	3780	29.3	219	6415	510	-1084	6075
	11	91	77.0	77.0	77.2	75.7	74.8	.....	64.5	75.4	338	4045	28.2	219	6175	900	-930	6379
	12	95	77.0	77.0	77.0	76.0	74.5	74.5	64.5	75.3	343	4070	22.2	228	5060	950	-856	5388
	1 P. M.	98	76.0	75.8	76.0	76.7	75.0	.....	64.9	75.0	355	3935	13.0	237	3080	1110	-44	4380
	2	98	75.5	75.5	76.0	75.0	75.0	74.2	65.5	74.5	268	2648	4.7	223	1050	1300	-421	2163
	3	98.5	75.4	75.0	75.4	76.0	75.0	.....	66.5	74.5	280	2460	0	0	0	1340	74	1648
	4	97	74.5	74.1	74.8	75.3	75.0	74.5	65.0	73.5	270	2521	0	0	0	1500	307	2041
	5	.....	75.4	75.2	76.0	74.7	75.0	.....	67.2	75.0	159	1360	0	0	0	1460	-308	1386
West	7 A. M.	71.8[b]	74.5	73.8	74.1	76.0	75.0	.....	65.2	74.5	.....	.....	0	0	0	-350	782	658
	8	71.7	74.9	74.2	74.7	75.3	74.2	75.4	66.3	74.8	60	557	0	0	0	-280	210	156
	9	75.7	74.8	74.1	74.4	75.0	74.0	.....	66.2	74.5	76	688	0	0	0	210	156	592
	10	81.8	75.1	74.5	74.8	75.0	74.0	74.8	66.6	74.8	74	665	0	0	0	660	-88	799
	11	84.6	74.8	74.0	74.5	74.8	73.0	.....	62.9	75.0	0	0	0	0	0	800	-119	907
	12	89.0	74.9	73.9	74.5	75.0	73.0	74.0	62.0	74.1	108	1443	0.7	228	160	1100	-174	1312
	1 P. M.	94.3	75.2	74.0	75.2	75.5	72.8	.....	61.7	74.5	167	2365	11.0	237	2605	1200	-116	3915
	2	96.5	75.0	73.8	75.0	75.3	72.0	74.0	60.5	73.8	313	4615	22.7	223	5060	1320	-77	6529
	3	97.9	75.3	74.5	75.2	75.8	72.0	.....	61.0	74.0	373	5375	33.0	200	6600	1360	-181	8005
	4	99.7	75.4	74.5	75.1	75.8	71.8	74.0	59.0	73.8	373	6120	31.0	183	5675	1440	-251	6868
	5	94.4	74.8	73.8	74.5	75.0	72.0	.....	62.1	74.1	132	1775	0	0	0	1400	-128	1469
North	7 A. M.	[a]	75.0	75.0	75.0	76.0	75.5	.....	74.0	75.5	.....	.....	0	.....	0	-370	322	185
	8	.....	75.0	75.0	75.0	75.5	75.5	.....	73.0	75.0	54	116	0	.....	0	-300	251	184
	9	77	75.0	75.0	75.0	75.3	75.5	.....	72.5	75.0	54	145	0	.....	0	210	108	551
	10	81	75.5	75.0	75.0	75.3	75.5	.....	72.5	75.0	54	145	0	.....	0	680	-34	879
	11	84	75.5	75.0	75.0	75.3	75.5	.....	72.5	75.0	94	255	0	.....	0	760	-34	959
	12	87	75.2	75.0	75.2	75.3	75.5	.....	72.2	75.0	111	335	0	.....	0	1100	36	1369
	1 P. M.	88	75.2	75.0	75.0	75.6	75.0	.....	71.2	75.0	132	545	0	.....	0	1190	71	1494
	2	90	75.0	75.0	75.0	75.0	75.0	.....	69.5	75.0	139	834	0	.....	0	1340	0	1573
	3	92	75.2	75.0	75.0	74.5	75.0	.....	70.0	75.0	48	261	0	.....	0	1420	-200	1453
	4	94	75.5	75.0	75.0	75.5	75.0	.....	70.5	75.5	75	410	0	.....	0	1430	0	1663
	5	.....	75.0	75.0	75.0	75.5	75.0	.....	70.5	75.0	146	714	0	.....	0	1360	-143	1450

[a] Temperature taken 6 in. from wall. [b] Temperature taken 1 in. from wall. [c] Heat given off by observers assumed to be 100 Btu per hour. Heat entering through wall assumed approximately 125 Btu per hour in east and west rooms and 135 Btu per hour in north and south rooms.

13 show little change. Seasonal variations in the position of the sun do, however, have an effect on the solar radiation entering through the windows. On June 21 with the sun high in the sky the normal area of sunlight entering through a window is a minimum. Likewise the normal area is a maximum on December 21. It follows that flat roof surfaces of a building receive the maximum amount of radiation in June whereas walls and windows receive their maximum later in the year. At noon in the latitude of Detroit the amount of heat received by vertical and horizontal surfaces on June 22 and September 11 expressed as a percentage of that received by a surface normal to the sun's rays is as follows:

	Vertical Surface	*Horizontal Surface*
June 22	30.6%	95.1%
Sept. 11	60.8%	79.3%

Fig. 9 shows that solar radiation through the window glass accounts for the largest part of the cooling required in the test rooms. The north room which received no sun needed a very small amount of cooling compared with that required in the other rooms. The effect of sunshine is especially evident in the west room where the amount of cooling required increased rapidly shortly after the sun began to shine in through the windows. In the latter part of the afternoon when the building across the street shaded this room, it was necessary to reduce immediately the amount of cooling air to prevent the room temperature from falling.

The large amount of heat entering a building through the windows in the form of solar radiation suggests that it would be wise to use some type of awning or shield to prevent sunshine from reaching the window

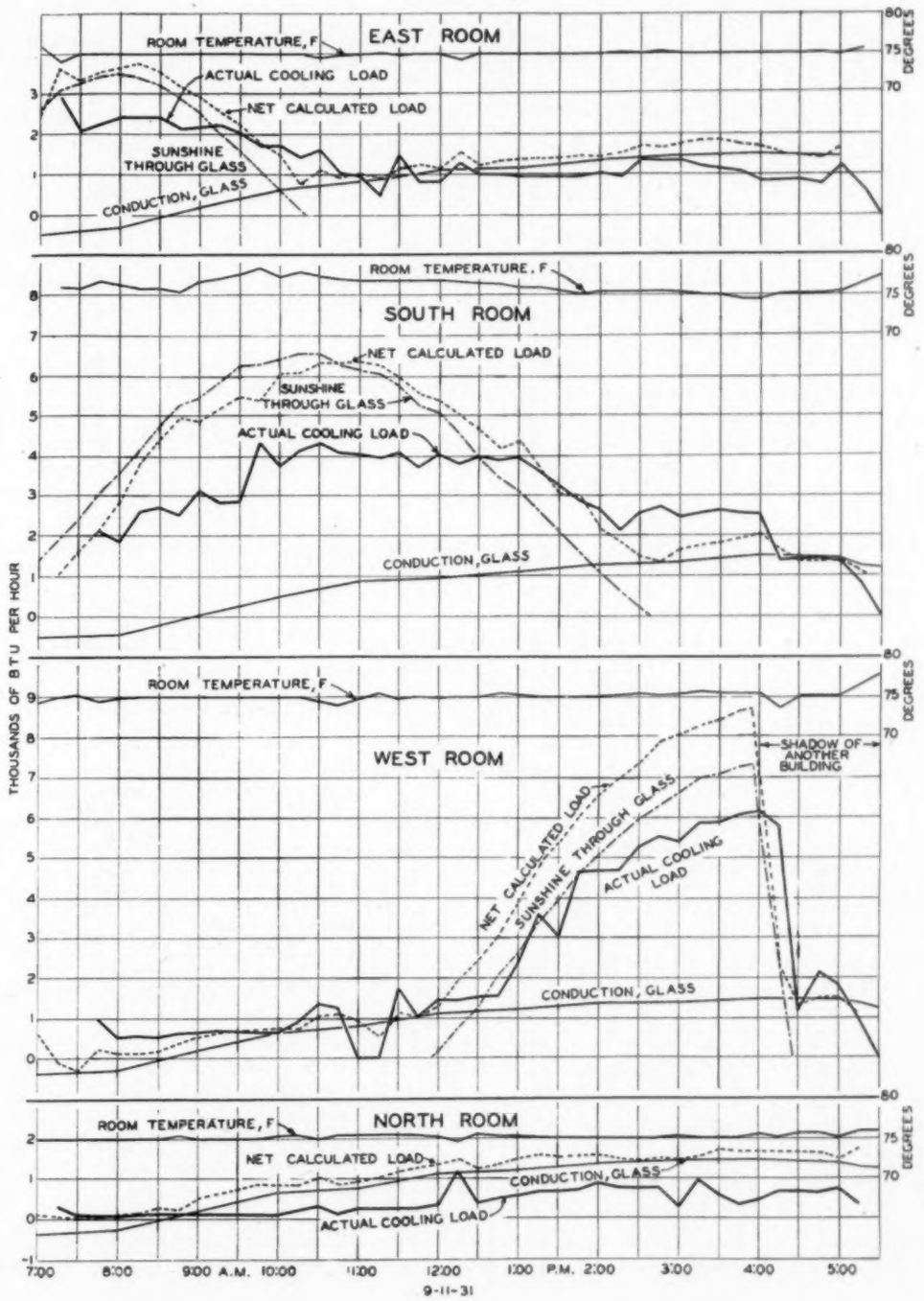


FIG. 9 — CALCULATED NET AMOUNT OF HEAT ENTERING THE FOUR TEST ROOMS COMPARED WITH THE ACTUAL AMOUNT REMOVED BY COOLING

glass. Venetian blinds and other types of inside window shades do not prevent solar radiation from entering through the glass although they doubtless do reflect some of it out again. Tests made on two successive days in the latter part of September, one day with the Venetian blinds covering the whole window and with the slats closed and the other day with the blinds up, entirely exposing the windows, showed that the rooms required as much cooling with the blinds as without them. It is possible that further tests would show that the blinds do make some difference but less than commonly supposed.

As a matter of interest to determine the absorption of solar radiation by double windows, pyrheliometer readings were taken through two panes of ¼-in. plate glass spaced 1¼-in. apart. The panes of glass were as clean as it was possible to make them. The absorption was 16.7 per cent through one pane of glass and 37.5 per cent through two panes. This indicates that double windows are of value in the summer in reducing the solar radiation entering through the windows as well as in reducing the amount of heat entering by conduction.

In all test rooms the actual amount of cooling required was less than the net calculated amount of heat entering the test rooms. In general the discrepancy was greater when the sun was shining in through the windows than at other times indicating that a large part of the error was in calculating the solar radiation delivered to the room. There are several possible explanations for this:

1. The window glass in the test rooms was not perfectly clean, whereas pyrheliometer readings taken to determine the absorption of solar radiation were made with clean glass. It is thus possible that the assumed loss in solar radiation in passing through the glass was too low and that the actual amount of heat delivered to the rooms was less than calculated.

2. During the tests the blinds were let down from the top as far as the meeting rail of the windows but were opened so that the slats lay in horizontal planes. This in general prevented the sunshine entering through the upper sash from striking the floor or walls of the room. The area of sunlight was calculated as though the blinds had not been there. It is possible that some of the heat was reflected out through the glass by them.

3. The temperature of the window glass is increased by the absorption of solar radiation and therefore the transmission of heat from the outdoor air through the glass is decreased and was probably less than calculated.

4. It is probable that part of the solar radiation striking the floor and interior walls was absorbed by them and transmitted to the spaces outside the room.

The lag in the cooling load in the south test room after the sun stopped shining through the windows was doubtless due in part to the heat being given up to the room by the floor and walls on which the sun had been shining. It will be noted from Fig. 9 that in the south room the temperature was above 75 F during the time the sun was shining in through the windows. This is because the part of the system serving this room did not deliver air of a sufficiently low temperature nor enough air to cool this room to 75 F.

The air quantities and temperatures are given in Table 3. The lowest air inlet temperature for the day in this room was 64.5 F whereas in the west room a temperature of 59 F was reached. The fan for the south end of the building ran at full speed from 9:30 a. m. to 1 p. m. but the air pressure in the supply duct was not great enough to supply sufficient air through one diffuser opening to handle the cooling load. If a temperature of 75 F had been maintained in the south room the actual cooling load would have been considerably higher.

The heat transmitted through the glass by conduction was calculated from the Weather Bureau temperatures. The first column in Table 3 indicates that the temperature varies on the different sides of the building but these temperatures were not considered sufficiently accurate to use in the calculations.

The calculated heat flow through the outside wall into the test rooms based entirely on inside and outside temperatures varied from approximately –80 Btu per hour (flow outward) at 7 a. m. to + 315 Btu per hour at 4 p. m. when the outside temperature was a maximum. Such calculations assume that equilibrium has been established through the wall. This assumption is, of course, incorrect in this case because the temperatures do not remain constant long enough to establish equilibrium. In view of this, the assumption used in this paper of 125 to 135 Btu per hour through the outside wall of the test rooms based on Nicholls' heat meter readings seems reasonable. At any rate, the heat entering through the wall is small compared with that through the windows, the calculated transmission through a test room wall being approximately 16 Btu per hour per degree difference and that through the windows 75 Btu per hour per degree. Just before 4 p. m. when the solar radiation entering the west test room was at a maximum, the heat through the wall accounted for only about 2 per cent of the heat being removed from the room by the cooling air.

As shown in Fig. 5, the test rooms were about 2 ft 4 in. narrower than a full bay. Because the amount of heat entering through the wall is so small, however, compared with the total, the results shown in Table 3 and Fig. 9 may be taken as substantially correct for a full bay. While the total amount of heat removed from the test rooms may be assumed the same as that from a standard office (neglecting lights and people), a lower rate of air renewal would be required in a standard office because of its greater width and depth. In the west test room under peak conditions air had to be supplied at the rate of about 375 cfm and at a temperature of 59 F in order to maintain a room temperature of 75 F. This amount of air is equivalent to six changes per hour in the office which would occupy the entire bay.

The results obtained in the test rooms cannot be considered typical of the total load of the building because of several factors:

1. Whereas the temperature of the air inside the test rooms was kept fairly constant, the inside temperature of the building as a whole varies with the outdoor temperature, being permitted to rise as high as 79 F during the hottest weather.

2. The test rooms had no lights and practically no occupants, while the building load was increased by the removal of heat and moisture from the occupants, heat and moisture from outdoor air taken into the system and heat from electric lights, fans, pumps and miscellaneous motors. The amount of outdoor air was 10 per cent of the total amount circulated except when the outdoor wet-bulb temperature was below 60 F when the amount of fresh air was increased to 50 per cent of the total and except

also for an outdoor temperature of over 92 F when the fresh air intake was often closed.

3. Two basements were air conditioned, which were unaffected by outdoor temperature or sunshine.

Measurement of Heat Flow Through Wall

The heat flow into and out of a section of the wall on the south exposure of the building was studied by means of two Nicholls' heat flow meters[6] placed directly opposite each other on the two surfaces of the wall. The wall at this point was made up as follows starting at the interior surface: plaster, 1¼-in.; tile, 3¾-in.; air space, 5¼-in.; tile, 3¾-in.; and brick, 9 in. The construction of the walls between the windows in the test rooms differed in that two courses of brick on the exterior were backed up solidly with brick and tile and one inch of plaster with no air space in the wall. The Nicholls' meter is particularly adapted to measuring the heat flow through walls of low conductivity and it is probable that the error introduced by the insulating effect of the meter plates themselves is small. There is an error, difficult to determine, caused by the different surfaces of the meter plate and the brick wall, an error which was probably greater in the case of the outside meter than the inside meter because the amount of absorption of radiant heat from the sun was increased due to the black surface of the meters. This is indicated by the fact that the temperature beneath the surface of the meter was slightly higher than the temperature of the surrounding wall surface.

Fig. 10 shows clearly how little of the heat entering the outer surface gets through the wall into the room. The sun began striking this wall at about 10:15 a. m. so that from that time on the heat absorbed by the wall was much greater than if the wall had been on the shady side of the building.

During the 24 hours from midnight to midnight on Sept. 11 the amount of heat passing through the two meters was as follows:

Heat entering outer surface of wall	243 Btu per square foot
Heat given up by wall to outside air	135 Btu per square foot
Net heat entering outer surface	108 Btu per square foot
Heat delivered by wall to room	29 Btu per square foot
Heat delivered by room to wall	3 Btu per square foot
Net heat delivered to room	26 Btu per square foot

This leaves 82 Btu per square foot entering the outer surface which did not appear at the inner surface Part of this doubtless was carried off through the air space in the center of the wall and the balance doubtless went to increase the temperature of the wall. Fig. 11 shows some evidence of this as the temperature of the inner surface of the wall increased from day to day and the temperature of the outer surface was also higher each night. Calculations indicate that each square foot of wall would require 28 Btu to raise its temperature only one degree.

A study of Figs. 10 and 11 indicates that it takes many hours for heat to pass in through the wall. The maximum absorption by the outer surface occurs from noon to 2 p. m., whereas during the night the inner surface of the wall delivers heat to the room at an increasing rate up to 7 a. m. when the air-conditioning system is started. The temperature of the inner surface of the wall rises during the night above the temperature of the air in the room indicating that this surface must be receiving heat from the interior of the wall.

[6] The Nicholls' heat flow meter is described in the paper, Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls. (A. S. H. V. E. Transactions, Vol. 30, 1924, page 65). It consists of a thin plate of formica with a series of thermocouples on each surface. The meter is calibrated in the laboratory by determining the relation between heat flow through it and the temperature difference between the two series of thermocouples, also taking into account the surface temperature of the meter. It is fastened to the surface of the wall and the heat flow through the meter, as indicated by the thermocouple readings, is obviously equal to the heat entering or leaving the wall surface.

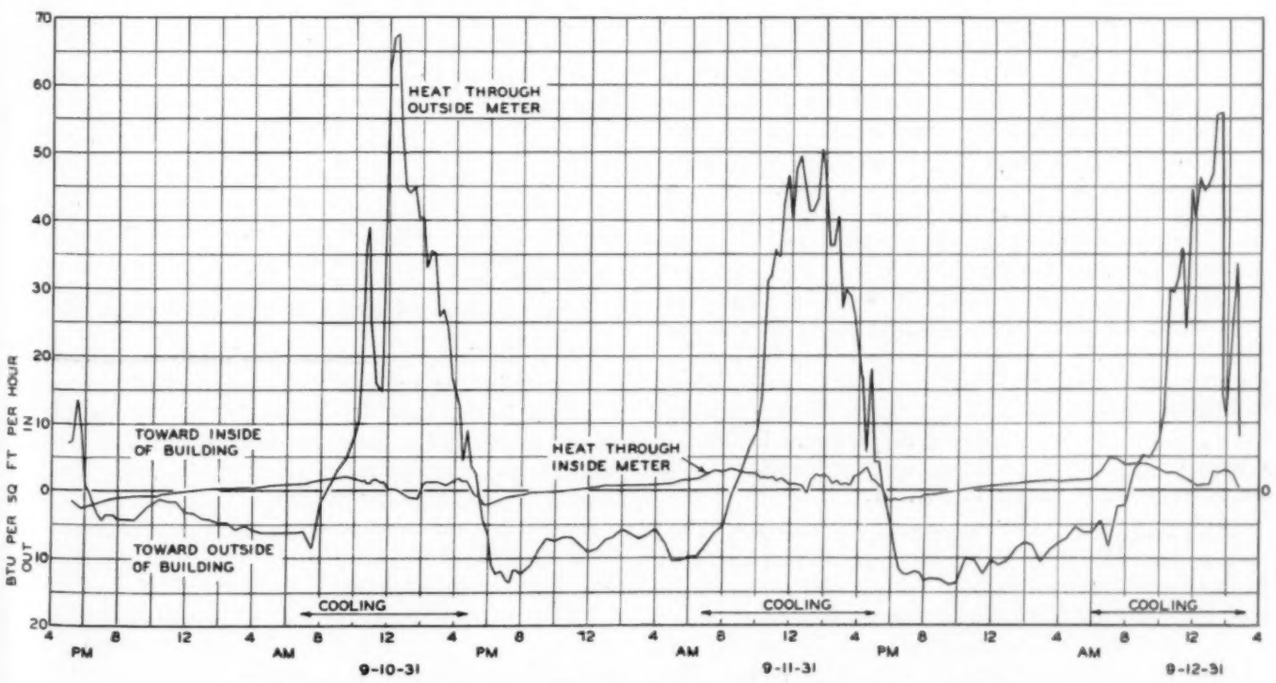


Fig. 10—Heat Flow Through Wall as Measured by Nicholls' Heat Meters

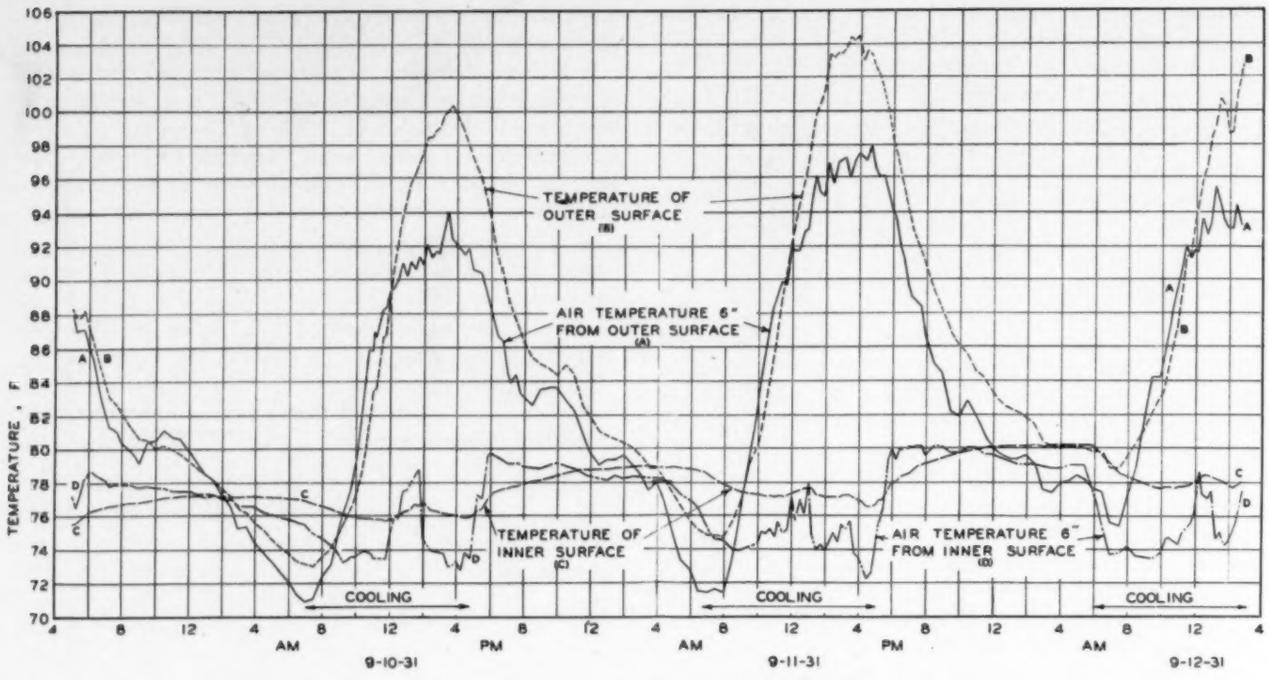


FIG. 11—WALL TEMPERATURES DURING HEAT METER TESTS

Fig. 11 shows the effect of the windows on room temperature. At 5 p. m. when the air-conditioning system is shut down and when the outdoor temperature is still high the indoor temperature rises rapidly to a point above the temperature of the inner wall surface, then drops during the night to a point below the temperature of the inner surface of the wall as the outside temperature drops. Thus the outdoor temperature acting through the window glass swings the indoor temperature up and down somewhat independently of the wall temperature.

It frequently occurs that heat is flowing into a wall from both surfaces at the same time or flowing out of a wall from both surfaces at the same time. Such conditions are indicated in Fig. 10. The wall gives up heat to the outdoor air all night but after midnight when the room is cooler than the inner surface of the wall, heat is also being delivered to the room. Likewise during the day, if the indoor temperature is allowed to rise above the temperature of the inner surface, heat will flow into the wall from the room at the same time that the outer surface is absorbing heat. This occurred during the afternoon of Sept. 10 when the room temperature in the place where the heat meter was located rose to nearly 79 F before it was checked by opening the damper in the ceiling outlet. It was more difficult to control room temperature by adjusting the standard ceiling outlet than by adjusting the dampers in the test rooms. This partly accounts for the fluctuations in room temperature shown in Fig. 11.

Table 4 gives wall and floor temperatures taken in the west test room. It is rather remarkable that the wide fluctuations in the temperature of the outer part of the wall have so little effect on the temperature of the room side of the wall. The temperature 1 in. beneath the interior surface varied only 2 deg in 24 hours. The temperature of the steel column is even more stable, varying only 1 deg in 24 hours. In the floor at a point 1 in. beneath the surface, the 24-hour variation was ½ deg. At a point 8 in. deep, the variation was 1 deg. All of this shows that the temperature of the interior building structure is very stable and that if it were not for the disturbance caused by heat entering through the window glass, temperature regulation in the summer would be a simple matter in an air-conditioned building.

Another striking fact is the utter insignificance of the wall conduction in the total cooling and dehumidifying load. Fig. 12 shows the relative magnitude of the various components of the cooling and dehumidifying load as calculated for a typical bay having the exposure of the south test room at a time when the sun radiation effect is near its maximum. The following assumptions were made:

Indoor dry bulb, 75 F; wet bulb, 62 F.
Outdoor dry bulb, 85 F; wet bulb, 72 F.
Air supply to room, 500 cfm.
Make-up, 10 per cent of air supplied.
Normal occupancy of room, 5 people.
Two 200-watt lamps, two more not in use.
Power for fans and pumps, average per bay for building.

Of all the factors, the heat entering through the wall is of least importance while the solar radiation through the glass is the most important factor. It should be pointed out that solar radiation through windows is not a constant and that at other times during the day the amount is less than shown here. The heat from electrical power used by fans and pumps was obtained by assuming that 90 per cent of the heat equivalent of the electricity becomes a part of the cooling load.

All of the data shown in Fig. 12 should not be taken as the average for the building because the solar radiation, number of lights, and number of people will vary in different parts of the building. Also there are other

TABLE 4—WALL AND FLOOR TEMPERATURES, WEST TEST ROOM, SEPTEMBER 11, 1931

TIME	AIR, 1 IN. FROM EXTERIOR SURFACE	WALL TEMPERATURES, DEG FAHR				AIR, 1 IN. FROM INTERIOR SURFACE	STEEL COLUMN	FLOOR TEMPERATURES, DEG FAHR	
		EXTERIOR SURFACE	1 IN. BENEATH EXTERIOR SURFACE	1 IN. BENEATH INTERIOR SURFACE	INTERIOR SURFACE			1 IN. DEEP	8 IN. DEEP
1 A. M.	80.0	81.1	83.0	79.4	78.9	77.4	76.6	75.2	74.7
2	79.2	80.4	82.4	79.3	78.9	77.3	76.6	75.2	74.8
3	77.6	79.4	81.8	79.4	78.9	77.2	76.8	75.2	74.9
4	77.4	78.6	80.7	79.3	78.9	77.2	76.8	75.2	74.9
5	74.4	76.2	79.8	79.3	78.8	77.1	76.9	75.2	75.0
6	72.5	75.7	78.5	79.2	78.8	76.8	77.0	75.4	75.0
7	71.8	74.9	77.4	79.2	78.3	75.5	77.0	75.5	75.2
8	71.7	74.4	76.6	78.7	77.8	75.2	76.6	75.2	74.8
9	75.7	76.7	77.7	78.2	77.4	75.3	76.7	75.2	74.8
10	81.8	80.7	79.7	78.6	77.8	75.5	77.0	75.3	75.0
11	84.6	82.7	81.8	78.5	77.9	75.8	77.4	75.2	74.9
12	89.0	86.8	84.6	78.3	77.5	74.7	77.3	75.1	74.8
1 P. M.	94.3	93.4	89.7	77.9	77.5	75.3	77.2	75.2	74.8
2	96.5	100.2	95.5	78.0	77.3	73.4	77.0	75.2	74.4
3	97.9	104.8	100.2	78.1	76.6	73.4	77.0	75.1	74.5
4	99.7	108.0	104.4	77.9	76.5	72.4	77.0	75.0	74.5
5	94.4	98.8	99.7	78.2	76.9	75.6	77.2	75.1	74.2
6	93.0	94.6	96.4	77.9	77.5	77.8	77.2	75.1	74.2
7	89.8	91.9	93.9	78.3	78.2	77.9	77.4	75.1	74.3
8	87.2	89.4	91.5	78.7	78.3	78.0	77.4	75.2	74.4
9	85.3	87.6	89.6	79.2	78.7	78.2	77.4	75.2	74.6
10	82.9	85.7	87.7	79.5	79.0	78.2	77.5	75.3	74.8
11	81.9	84.3	86.4	79.7	79.2	78.3	77.5	75.4	74.8
12	80.2	83.1	85.3	79.9	79.3	78.3	77.6	75.5	75.0

sources of heat not listed here such as infiltration, hot water piping, and miscellaneous electric motors.

Total Cooling Load of the Building

Another phase of the work was a study of the total cooling and dehumidifying load of the entire system to determine its variation with outdoor temperature and amount of sunshine. The cooling and dehumidifying load was determined by subtracting from the heat removed by the condenser, the heat equivalent of the electrical power used to drive the compressors. This gave a relatively close estimate of the load, but it cannot be considered to be entirely accurate.

The curves in Fig. 13 show the seasonal variation of cooling load, plotted with outdoor temperature and the amount of sunshine. The outdoor temperatures given are the average of the hourly readings of the U. S. Weather Bureau from midnight to midnight. Sunshine is

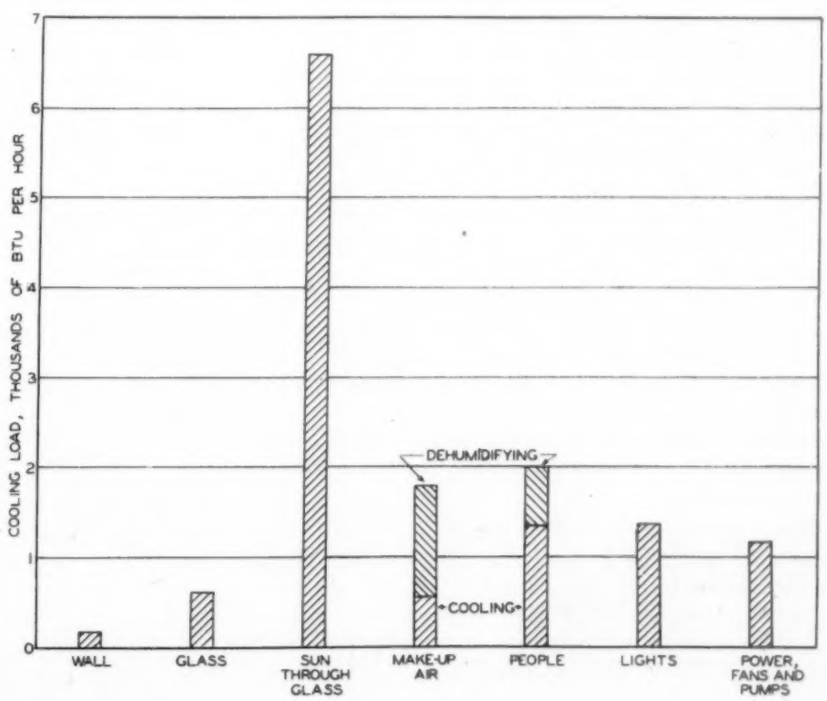


FIG. 12—DIVISION OF COOLING AND DEHUMIDIFYING LOAD, TYPICAL SOUTH BAY

expressed as the number of hours per day that the sun cast a distinct shadow. This does not give an accurate measure of solar radiant heat because the heat intensity varies with the time of day. On Monday evenings parts of the air-conditioned space are in use so the cooling system is operated for longer hours than on other days. In Fig. 13 the points surrounded by circles represent the actual amount of cooling on Mondays. In order to make the data comparable with those on other days, the cooling on Mondays was reduced by the number of extra hours of operation on Mondays. The points thus obtained were plotted as a part of the curve. Also on Saturday afternoons the cooling system is shut down earlier than on other days. The points plotted as crosses represent the actual amount of cooling on Saturdays. This amount was corrected by increasing it in proportion to the longer hours of operation on other days and the corrected points were plotted as a part of the curve.

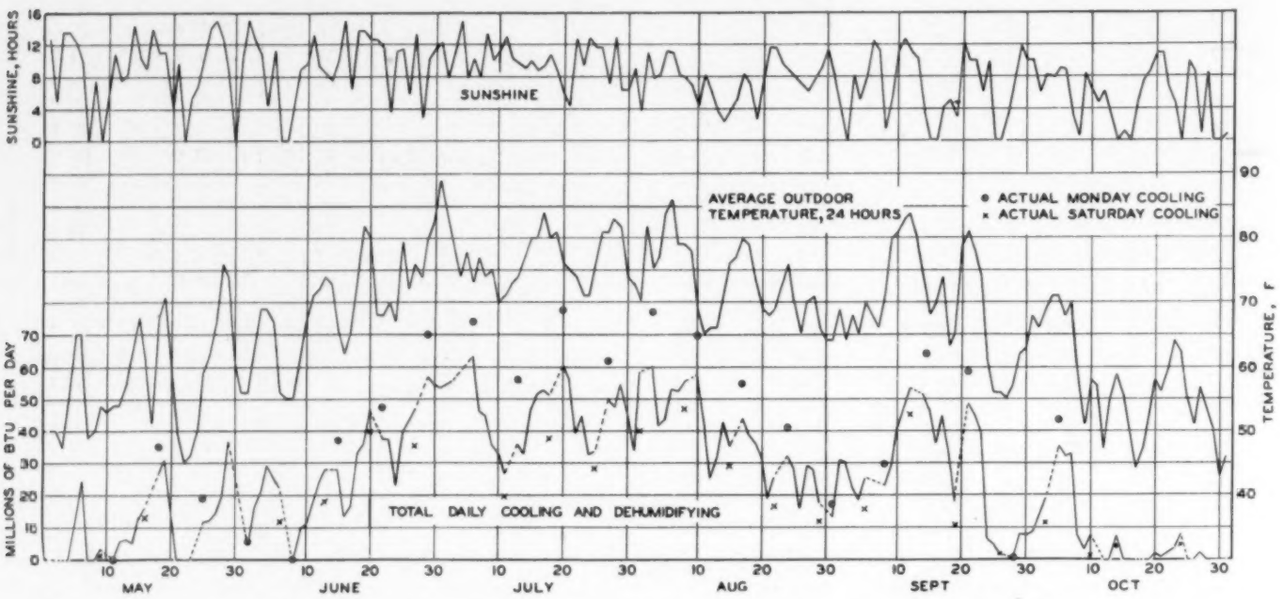


FIG. 13—TOTAL DAILY COOLING AND DEHUMIDIFYING BY THE AIR-CONDITIONING SYSTEM, SUMMER OF 1931

During the period May to October, inclusive, the refrigerating apparatus was in operation for 1,203 hours. Of this time there were 98 hours when all three compressors were in use. In other words, during the hours of operation it was necessary to use three compressors about 8 per cent of the time.

The curves in Fig. 14 show the hourly variation of cooling load plotted with the outdoor wet- and dry-bulb temperatures and the amount of sunshine. Dry-bulb temperatures were obtained from the U. S. Weather Bureau, as well as the hourly per cent of sunshine. Wet-bulb temperatures were taken from the records of the Union Guardian Building operating engineers and were measured in an air intake.

The curves in Fig. 13 show a noticeable correlation between the total daily cooling load and the average daily outdoor temperature. This, however, does not signify that heat transmission through windows and exterior walls is the primary cooling factor, because high dry-bulb temperatures usually are accompanied by increased solar radiation and high wet-bulb temperatures, factors which add to the cooling load in a different manner.

It should be noticed that variations in cooling load lag slightly behind the variations in average daily outdoor temperature. The explanation of this lag can partially be understood from Fig. 14. It is apparent that the operation of the cooling system is stopped in the afternoon at a time when the outside temperature has usually reached its peak, leaving many hours of high temperature and sunshine during the remainder of the day to add heat to the building, particularly on the west side of the building. Much of this heat is dissipated outward through the windows if the outside temperature falls below the inside temperature during the night.

Two other factors operate to heat the building during the evening; one is exfiltration which takes place during a warm evening due to the vertical column of relatively cool air inside the building, 16 stories in height. The second factor is addition of heat from lights used by janitors during the evening, when no fans are being operated. During the hours after midnight heat also enters the building from the exterior walls, as shown in Fig. 10. The result is that the cooling system must be started from one to three hours before the offices are occupied, and cooling continued at a fairly high level during the early forenoon to care for the heat absorbed by the building and furniture.

Sunshine, which greatly affects the cooling load of a particular room, does not affect the total load of the building to the same degree because only a portion of the building is exposed to the sun's rays at a time. Sunshine produces its greatest effect on the Union Guardian Building after 1 p. m. This is due to the orientation of the building, which is such that the east side gets the morning sun at a sharp angle, letting a relatively small amount of low intensity sunlight into the windows. The south side of the building is narrow, so that although the sunshine strikes this side squarely and brightly, only a small

percentage of the total exterior surface of the building is affected. Sunlight enters the west windows after noon and a large expanse of wall and window surface is then affected by solar radiation of high intensity, except that the north half of this side is gradually shaded by a neighboring building toward the end of the afternoon.

Referring to Fig. 14, sharp increases in the total cooling load can be seen to have occurred on the afternoons of June 24 and 25, days when the sunshine was bright during the entire cooling period. The effect of heat entering the building at night, or rather the lack of heat dissipation at night, can be seen in the cooling loads of June 26 and 27. The load on June 26 between 8 a. m. and 12 noon was higher than the load on June 23 for the same period, although the dry-bulb and wet-bulb temperatures as well as the sunshine averaged somewhat lower. Comparing the same hours on June 27 with those on June 25, it can be seen that although the temperatures and sunshine are comparable, the load on June 27 is the higher, presumably because of the warmer preceding night.

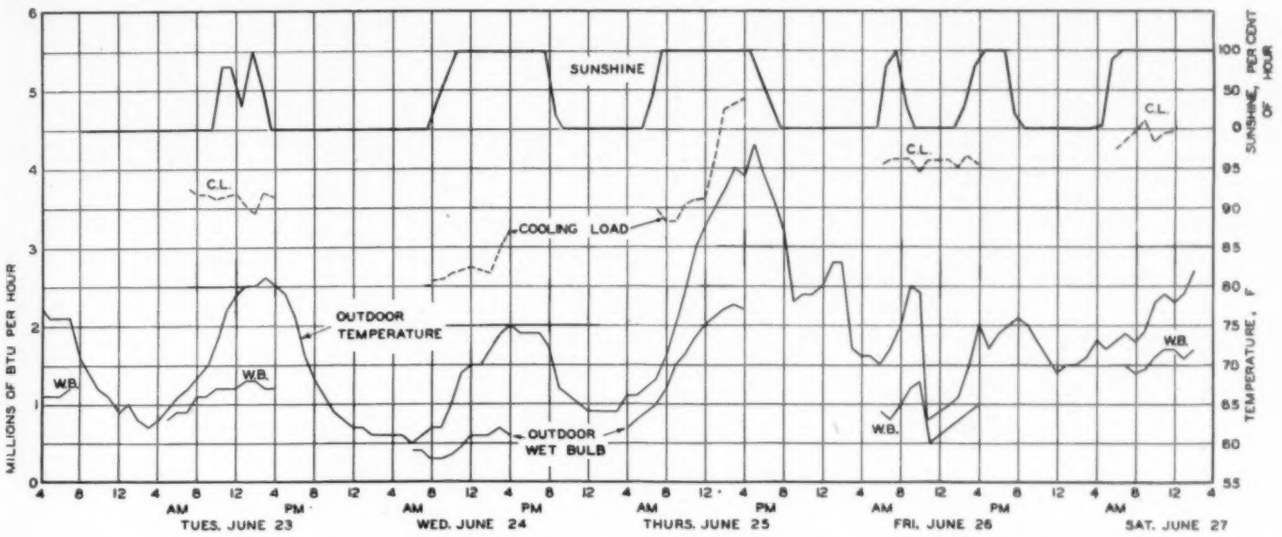


FIG. 14—VARIATION OF BUILDING COOLING LOAD WITH OUTDOOR TEMPERATURE AND SUNSHINE, JUNE 1931

Conclusions

Sunshine through the window glass is apparently the most important factor to contend with in the cooling of a room. At times it may account for as much as 75 per cent of the total cooling necessary. Because of the importance of the sunshine, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. Consideration should also be given to shadows from nearby buildings which may cover part of the sunny side of a building. If buildings were provided with awnings so that the window glass were shielded from sunshine, the amount of cooling required would be reduced and there would also be less difference in the cooling requirements of different sides of the building.

In modern office buildings in the latitude of Detroit the amount of heat entering through the walls in summer is of little importance compared with that coming through the windows. It appears that the use of double windows would reduce the amount entering by direct radiation from the sun as well as that by conduction.

A study of the cooling for the entire building from Fig. 13 indicates that the total amount of cooling for any day (Mondays and days following holidays excluded) is in general proportional to the difference between the average temperature for the day and 55 F. It is affected somewhat, however, by the outdoor conditions prevailing during the previous day.

In the latitude of Detroit appropriate design figures for the heat entering the walls and windows of an office building are as follows:

Wall transmission...	3 Btu per hour per square foot of wall
Glass transmission..	17 Btu per hour per square foot of glass
Sunshine (east and west sides).....	160 Btu per hour per square foot of glass
Sunshine (south side)	140 Btu per hour per square foot of glass

The maximum rate at which solar radiation enters south windows is less than for east and west windows because of the angle at which the sunshine passes through the glass. The glass area referred to is the area actually exposed to the sunshine. If there are overhanging ledges above the windows, these must be taken into account.

Acknowledgments

The Nicholls' heat meters and the Abbot pyrheliometer used in the tests were loaned by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Acknowledgment is due F. C. Houghten for his helpful suggestions and comments, to the Union Guardian Trust Company which cooperated in making the tests, and to C. B. Sprenger and C. W. Signor of The Detroit Edison Company who conducted the tests and contributed many suggestions.

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A Study of Intermittent Operation of Oil Burners

By L. E. Seeley[1] (*MEMBER*) and J. H. Powers[2] (*NON-MEMBER*), New Haven, Conn.

This paper is the result of research conducted at Yale University in cooperation with the A. S. H. V. E. Research Laboratory and the American Oil Burner Association

INTERMITTENT operation of an oil burner refers to its occasional use in order to vary the average heat output required by heating practice. The greater the proportion of time that the burner is idle the lower will be the average heat output of the boiler in which it is installed and vice versa. The maximum heat output will be secured when the burner operates continuously and in practice this output is appreciably greater than the maximum heat loss of the structure to be heated. It is evident that a burner cannot operate or be *on* for any considerable period of time.

It is therefore a characteristic that most oil-fired heating systems operate intermittently and the question arises as to the effect of this on the economy of operation and also whether or not there is a preferred method of intermittent operation control. A comprehensive study should consider other methods of achieving the variable heat output required and their influence on economy of operation.

There are two other methods, namely the *continuous* and the *high and low*, the manual control not being considered because of its unpredictability in actual practice. The *continuous* method refers to those burners which operate steadily throughout the heating season and obtain variable heat output by automatically changing the rate of fuel burning or size of flame to suit the immediate requirements. The *high and low* method refers to a continuously operating burner which fluctuates from a low to a high flame and vice versa, the average heat required being obtained by regulating the duration of the *high* flame. The most obvious difference between these two methods and the one first mentioned is that there is a certain minimum output of heat that cannot be avoided, whereas the *on and off* burner may vary the heat output from zero to its maximum. The study of these methods will be confined largely to matters of economy or efficiency of operation although there are undoubtedly other operating features equally worthy of study.

The A.S.H.V.E. Proposed Code for Testing Steam Heating Boilers Burning Oil Fuel[3] provides for the making of intermittent tests. This was necessary in order to obtain an idea as to what might be expected from the equipment as it is actually used. An efficiency curve obtained by the intermittent burner operation will show the economy that may be expected; hence comparisons of such efficiency curves should be the most enlightening.

Intermittent and Continuous Efficiency Curves

It was shown in a previous paper[4] that intermittent efficiency curves have a definite character although the curves obtained by continuous tests at various fuel rates do not. Fig. 1 shows two curves, one by intermittent and the other by continuous tests as provided for in the test code. The extreme righthand end of the intermittent curve represents the output and efficiency obtained when the burner is operating continuously at a

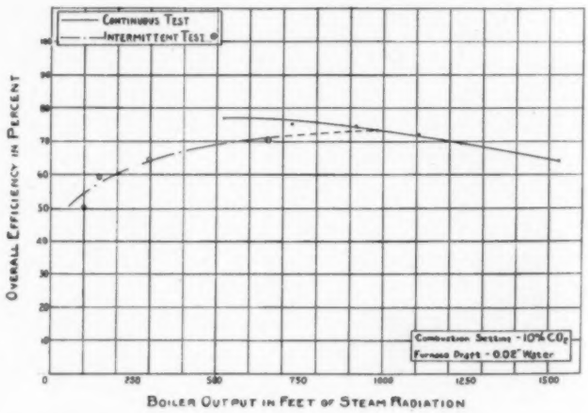

FIG. 1—CONTINUOUS AND INTERMITTENT TEST EFFICIENCY CURVES

uniform fuel-burning rate. This point would naturally lie on the other curve, since it was obtained from a continuous test. The other values along the curve are obtained by operating the burner only a portion of time during the test period.

The intermittent curves of Fig. 1 were obtained by setting the burners to operate for 30 min under definite conditions of fuel rate, draft and excess air, followed by *off* periods of different durations for different tests. Comments on this procedure indicated a desire to know what the results would have been with *on* or operating periods of other than 30 min, the feeling being that while 30 min might be a reasonable period to choose it was still arbitrary and gave no assurance that the results were typical of actual conditions. Furthermore, it did

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[3] *Heating, Piping and Air Conditioning*, April 1931, p. 338.

[4] Study of the Characteristics of Oil Burners and Heating Boilers, by L. E. Seeley and E. J. Tavanlar (*Heating, Piping and Air Conditioning*, May 1931, p. 419).

Presented at the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1932.

not show what was to be expected if short *on* periods were used as might be the case if steam pressure rather than thermostatic regulation were employed.

Before replying to this question it would be advisable to ascertain just what the intermittent tests with the 30-min *on* period show. All of these tests were plotted by raising or lowering the curves so they would start at an output of 1000 sq ft[5] of equivalent radiation and 75 per cent efficiency, this being the extreme righthand point of the intermittent curve of Fig. 1. The object was to superimpose the curves to ascertain whether or not there was a resemblance in shape. This would show for every combination of boiler and oil burner what the intermittent efficiency curve would be if, during continuous operation, its efficiency were 75 per cent.

Fig. 2 shows a complete intermittent efficiency curve for *on and off* burners which was made possible because all of the curves came so close together. It can be stated with reasonable certainty that all *on and off* oil burners adjusted to the same furnace draft and same percentage of excess air which can give the same boiler output and

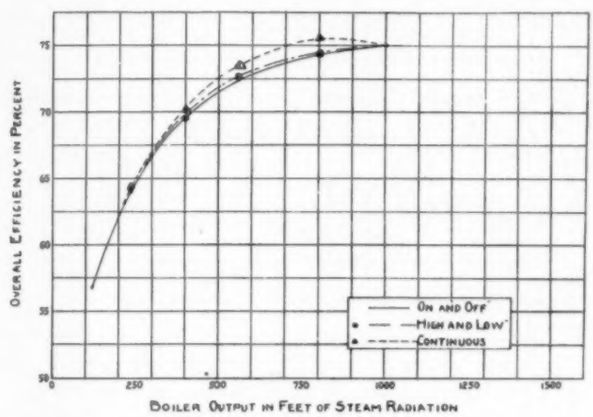


Fig. 2—On and Off, High and Low, and Continuous Test Efficiency Curves

efficiency when *full on* will produce practically identical results on intermittent operation. Likewise all boilers that will give the same output and efficiency with *full on* burners similarly adjusted as to draft and excess air should produce practically identical economy in actual heating use. The fact that the shapes of all intermittent efficiency curves are nearly the same is the only factor to consider in Fig. 2. The efficiency values should be disregarded since these will vary above or below but parallel to the curve shown according to the actual performance of each combination.

An average curve for *continuous* burners is also shown in Fig. 2. It should be noted that the curve is about the same as the other though slightly higher. However, the conclusions relative to operating economy previously expressed are not correct for this type of burner. If two boilers were each equipped with two *continuous* burners similarly adjusted and giving the same output and efficiency at maximum fuel rate, it does not follow that identical economies would result in actual use. Heat absorption characteristics of the boiler will have an influence which is totally lacking with the *on and off* burners. Figs. 3 and 4 will illustrate the reason.

[5] One square foot = 240 Btu per hour at 215 F (steam) and 70 F (air).

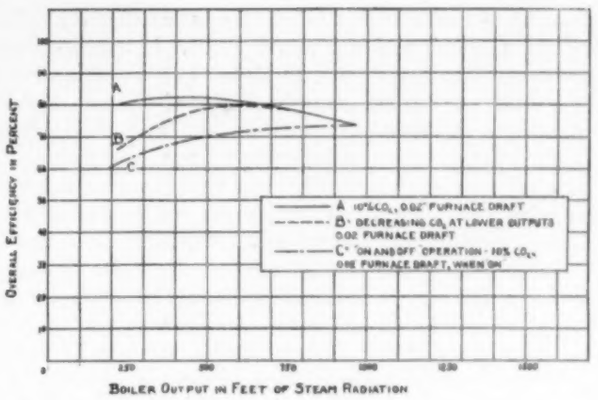


Fig. 3—Carbon Dioxide and Furnace Draft Efficiency Curves for Maximum Output Beyond the Maximum Efficiency of the Boiler

Influence of Boiler Characteristics

Fig. 3 shows that if maximum output is beyond the maximum efficiency of the boiler, the continuous burner will tend to follow the curve for a while and then fall away at low outputs while the *on and off* curve starts to fall away almost immediately. Fig. 4 shows that if the point of maximum output is at or near the maximum efficiency point of the boiler the continuous burner will tend to follow the curve as before and hence will drop downward in the same manner as the *on and off* curve.

Curves *A* and *D* represent the type found by continuous tests at various fuel rates when the quality of combustion and draft are maintained constant. Curves *B* and *E* are obtained by changing the fuel rate for each test. While the air supply is also automatically changed, the quality of combustion (*i. e.* per cent CO_2) does not remain constant as in the previous case and due to increased excess air the efficiency drops away from *A* and *D*, especially at the lower outputs. It should be noted that Curves *C* and *F* are the same in accordance with the conclusions relative to *on and off* burners. In actual practice comparisons between these two types of burners would be complicated by the fact that they would probably not be set for the same maximum output to take care of identical heating loads. Figs. 3 and 4 do not represent actual test curves but do represent characteristics found in tests and are merely combined to make

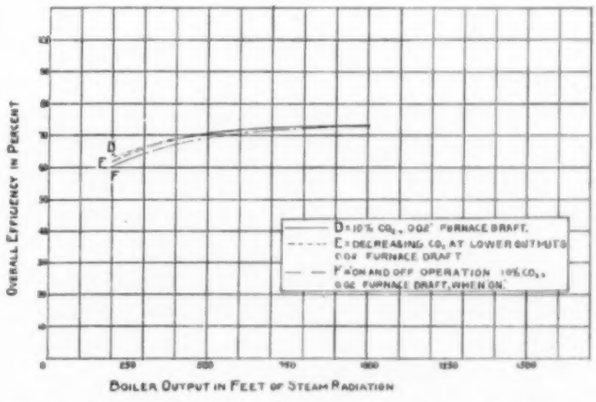


Fig. 4—Carbon Dioxide and Furnace Draft Efficiency Curves for Maximum Output at or Near Maximum Efficiency of Boiler

clear the distinction that is evident in actual tests. The boiler on which the curves of Fig. 3 is based should give better results with a continuous burner than with an *on and off* burner over the range of outputs shown. Low average outputs were not obtained and comparisons should not extend beyond the limits actually shown.

Fig. 2 also shows results that may be obtained with *high and low* burners. They are practically the same as those obtained from the *off and on* types. It shows in this case that the efficiency with the *low* flame is lower than that obtained with the *high* flame. It seems reasonable to suppose that the average efficiency due to *high and low* operation will lie between the two values, never higher and never lower. Improvement in average efficiency would result from a better *low* flame operation.

On Periods of Different Duration

With such a similarity shown for *on and off* efficiency curves where the *on* period used was 30 min, how will the economy be influenced by *on* periods of different duration? To answer this question, *on* periods of 5, 10, 15 and 30 min were selected and four *on and off* efficiency curves determined. *Off* periods were chosen to give the following time ratios[6]: 0.0, 0.2, 0.5, 1, 2 and 3.

Fig. 5 shows Series *ES1* tests which were obtained by setting the furnace draft at 0.02 in. of water and the combustion to show 10 per cent CO_2, while the burner was *on*. Automatic draft regulators were used. The efficiencies are not very different, all falling within the limit of probable testing error which leads to the conclusion that the 30-min *on* period tests were typical of intermittent tests in general. It would appear that steam-pressure or thermostatic regulation would be equally good.

Fig. 6 (*Series FS1*) shows the same result insofar as effect of timing periods is concerned. This series differs in one respect from *ES1*. The furnace draft was set at 0.07 instead of 0.02 in. of water; the CO_2 was kept at 10 per cent as before. Figs. 7 and 8 show the same efficiency curves plotted in Figs. 5 and 6 except that the ratio of the *off and on* periods is used instead of boiler output.

Effect of Furnace Draft

Comparison of Series *ES1* and *FS1* shown by average curves in Fig. 9 reveals that the average efficiency with higher drafter is not as good as *ES1* in spite of the fact that both *on* adjustments produced the same results during periods of burner operation. This suggests that

[6] Ratio $= \frac{\text{time off}}{\text{time on}}$

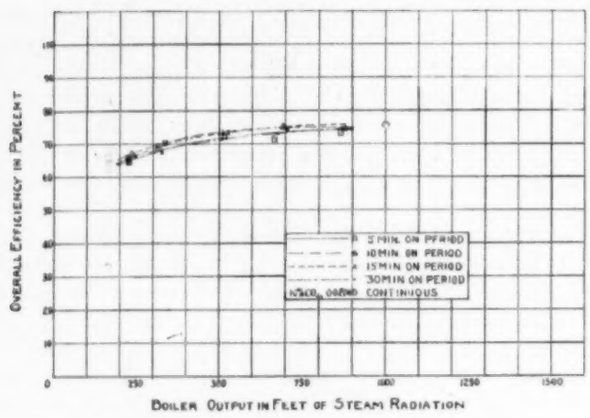


Fig. 5—Intermittent Efficiency Curves *ES1* Showing Relation Between Boiler Output and Overall Efficiency

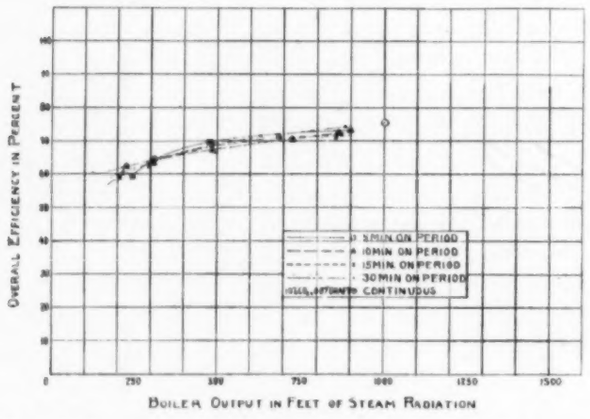


Fig. 6—Intermittent Efficiency Curves *FS1* Showing Relation Between Boiler Output and Overall Efficiency

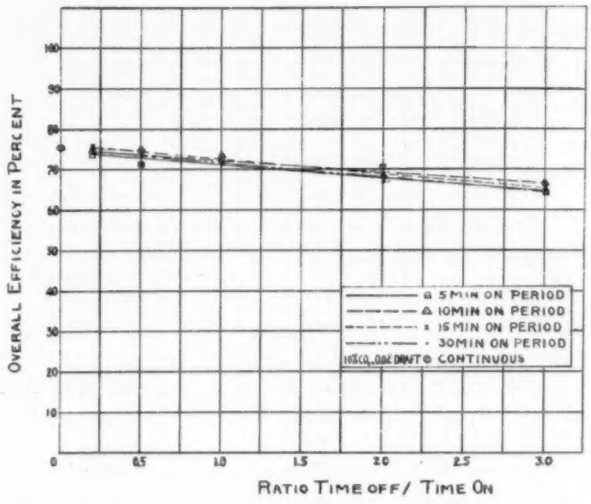


Fig. 7—Intermittent Efficiency Curves *ES1* Showing Relation Between Overall Efficiency and Ratio of Time Off to Time On

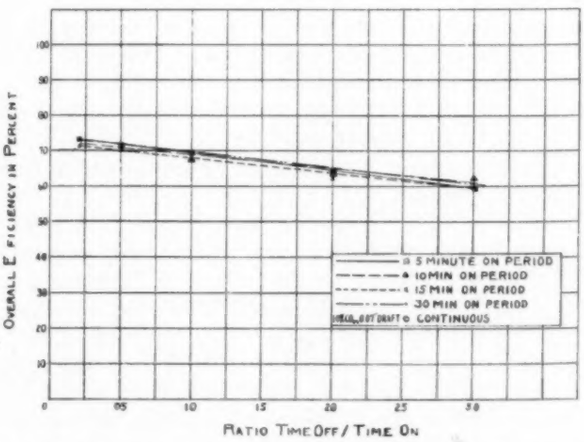


Fig. 8—Intermittent Efficiency Curves *FS1* Showing Relation Between Overall Efficiency and Ratio of Time Off to Time On

for best results burners should be adjusted for proper fuel rate and combustion with as low a furnace draft as advisable for the particular installation conditions. A furnace draft of 0.02 in. of water would probably be considered too low for *on and off* operation but 0.05 in. of water should prove satisfactory in the majority of cases. The exception may be found among those burners which do not supply all the air required for combustion by means of a fan. These burners should be adjusted for proper combustion first of all and the furnace draft to supply the required air may or may not equal the recommended value. Fig. 9 explains why the regulation of draft to similar values was one of the prerequisite conditions mentioned in the discussion of Fig. 2. If this were not done it would be possible to get two combinations of boilers and burners to produce identical results while *on* but if the furnace drafts were different the economy of intermittent operation would not agree.

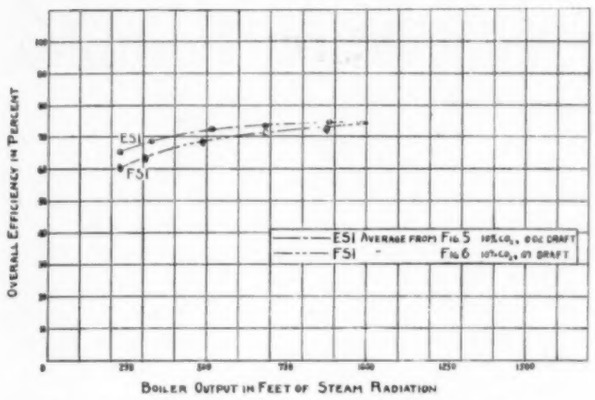


Fig. 9—Average Intermittent Efficiency Curves

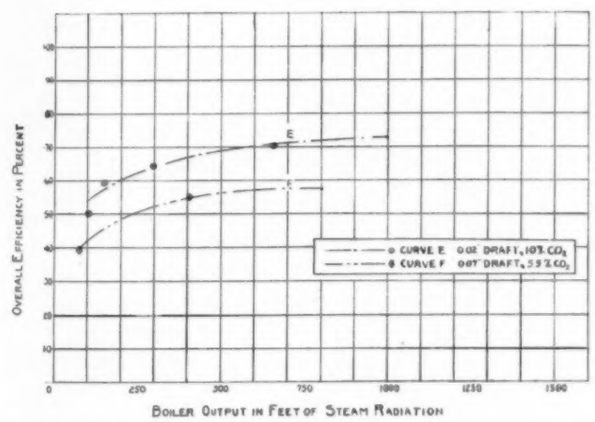


Fig. 10—Intermittent Efficiency Curves Showing Effect of Furnace Draft Fluctuations

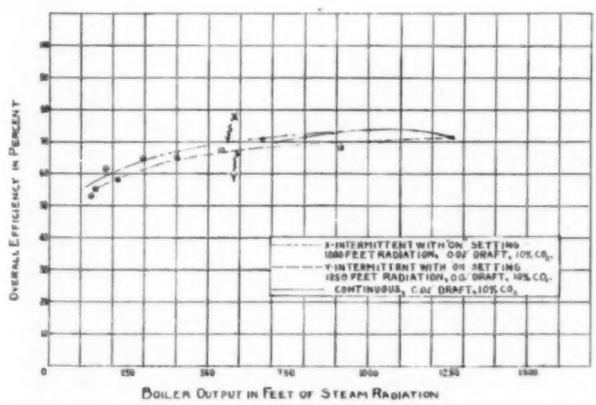

Overall Efficiency in Percent
X-Intermittent with On Setting
Y-Intermittent with On Setting
Continuous
Boiler Output in Feet of Steam Radiation

Fig. 11—Intermittent Efficiency Curves at Different Output Settings

The information given in Fig. 9 should not be confused with results reported in a previous[7] study showing the effect of draft fluctuations on burner performance. Those tests were obtained by adjusting the burner as described for Series *ES1*. The furnace draft was then increased to 0.07 in. of water but in distinction to Series *FS1* no air adjustments were made to secure 10 per cent CO_2. The combustion was allowed to go where it would. Fig. 10 shows the influence that draft fluctuations may have on the intermittent operation economy. While some *on and off* burners are more affected than others in this respect, it may be concluded that once the proper draft setting is made it should be maintained as constant as possible.

Setting of Burner

May one capacity setting for a combination of boiler and burner be better than any other from the standpoint of economy? Fig. 11 shows two intermittent efficiency curves. In one case the *on* setting was at a capacity of 1265 sq ft of radiation and 72 per cent efficiency and in the other 1000 sq ft of radiation and 74 per cent efficiency. It should be noted that the intermittent curve *x* is the higher of the two and its *on* period efficiency is higher. This would be expected from a study of Fig. 2. The best setting for economy in any case, therefore, is that operating capacity which gives the highest efficiency of the boiler in question. If this type of setting is properly proportioned to the heating load requirements of an actual building, the highest fuel economy should result.

It must be remembered that all tests referred to were made on the equivalent of an ordinary low-pressure steam system. It is not at all certain that a warm air, hot water or vapor system would show the same results because the heat could probably continue to flow into the system for a longer time during the *off* period and the loss of heat up the chimney might be reduced. There is no definite proof of this but plans are being made to investigate the question. Certain types of systems might prove more desirable than others from the restricted viewpoint of fuel economy to which this study has been confined.

Conclusions

The following conclusions seem to be justified from the information at hand:

1. The intermittent efficiency curve for *on and off* and its equivalent for other types of burners is the true indicator of fuel economy to be expected in actual practice.

2. All intermittent curves or equivalent have substantially the same characteristic shape for similarly adjusted burners.

3. Any number of boiler and burner combinations which have the same capacity, efficiency, furnace draft

[7] Study of the Characteristics of Oil Burners and Heating Boilers, by L. E. Seeley and E. J. Tavanlar (*Heating, Piping and Air Conditioning*, May 1931, p. 419).

and excess air or combustion setting while *on* will produce substantially identical results in actual use.

4. Continuous-type burners are influenced by the heat absorption efficiency of the boiler over the whole range of output.

5. *High and low* burners tend to follow the *on and off* type of efficiency curve. The curve should be confined between low and high limits of efficiency actually obtained by the *low* and the *high* flames respectively.

6. The lower the furnace draft setting and the more uniformly it is maintained, all other things being equal, the better the fuel economy.

7. Various *on* period timings seem to have practically no effect on the intermittent efficiencies.

8. The best economy on intermittent operation for any combination is an *on* setting which produces the highest efficiency.

9. It is possible that certain types of heating systems may be better adapted for economical intermittent operation than others but this is not definitely known.

Diagnosing Heating and Ventilating Troubles

By S. R. Lewis

The engineer who is called in to diagnose troubles with heating systems must be an experienced diagnostician. He needs, in many cases, to have intuition, because the basic reasons for trouble are not always easily located. Probably more heating-plant complaints are terminated by achieving cleanness than by means of any other prescription.

It is well known that the perfunctory washing out of core sand which sometimes occurs at the point of manufacture of ferrous castings does not by any means clean the castings. While any system of screwed piping is being fabricated there is a steady application of lubricating oil, and there are continual opportunities for cuttings, sand, plaster, mud, shavings, and other construction debris to enter the piping.

In case of trouble, experienced engineers almost invariably advise cleaning the piping and the heating units before looking elsewhere. Assistance may be given by the introduction into the boiler or system of chemicals which will neutralize some of the foreign elements, but eventually if the surfaces and pockets are clean, and the water is clean, the system will be free of unpleasant noises, and the fuel cost will be reduced.

Where excessive quantities of chemicals have been introduced it is not uncommon to find an acid condition in the condensate, and this may attack copper interiors, return traps and air vents, with the result that floats and expansion elements may develop leaks. Such leakage into any vacuum heating system of course prevents the maintenance of a proper vacuum and results in the necessity of the introduction of a jet of water at the pump. Where such a jet of water is introduced, raw water with its impurities usually must be supplied, calling for more chemicals which aggravate the situation. An examination of the interior parts of traps and air vents often reveals the seat of this sort of trouble.

The best test for water-cleanness usually is to put a sample of the water suspected of being polluted, in a dish on a stove alongside a similar dish containing fresh water. The fresh water usually will be boiling for some time before the higher surface tension of the polluted water can be broken.

When a boiler is blown off at its bottom the water line of course goes down, and much of the foreign matter, especially the oil, lighter in weight than water, which was floating above the water, clings to the boiler interior and contaminates the fresh water supply. The easiest way to get rid of this form of pollution is by the use of a blow-off opening at the water line which can operate as a skimmer. Many cases of supply valves to radiators, of return traps, and of pipes, all of which were suspected of being too small for the work which they were supposed to do, have been solved by a simple cleaning, without making any physical changes.

The removal of air and condensate from steam heated units, especially those in connection with a fan, is of major importance and manufacturers of such equipment offer suggestions in their literature as to the physical structure which should not be ignored. In many cases the real difficulty is by no means evident, and it is very easy to overlook a serious structural fault. In a large school where complaint was made that the ventilation, formerly excellent, recently had become defective, it was found that the cold air intake and the exhaust openings were functioning and that the thermostats were operating. They were operating the housed supply fan in the wrong direction! When such a fan is operated backward the amount of air delivered may be about half of that normally delivered, and as the power demand is proportional to the work done, the electric motor functions satisfactorily. In this case, a careless workman in reconnecting the motor had transposed two of the leads and no one had noticed, in the dark fan chamber, the fact that the rapidly-turning wheels were going in the wrong direction.

In another case of deficient air delivery, the motor pulley was ordered correctly as 8 in. in diameter and the shipping list of the manufacturer and all of the records showed an 8-in. pulley, but unfortunately when the manufacturer sent out the motor he placed a 6-in. diameter pulley on it. Naturally the fan which this motor was propelling failed to deliver as much air as was expected and several days' delay and a long railroad trip were expended before the difficulty could be diagnosed.

Occasionally a thermostat in a school classroom is connected by mistake so as to control the air supply damper of some other classroom. Such a cross connection is often a long time in being discovered and corrected and may lead to very interesting temperature-complications. Thus the trouble shooter learns to doubt everything he reads or hears and to be suspicious even of what he sees and of what he feels.

Transmission of Radiant Energy Through Glass

By R. A. Miller[1] (*MEMBER*) and L. V. Black[2] (*NON-MEMBER*), Pittsburgh, Pa.

THE object of this paper is to point out certain facts in connection with the transmission of radiant energy through glass, and if possible, to correct the somewhat prevalent idea that large radiation losses occur through the glass. It shou'd be borne in mind that there are two different forms of energy, namely:

1. Radiant energy, which is an electro-magnetic wave motion.
2. Sensible heat, which is molecular kinetic energy in matter.

Radiant energy is transformed into sensible heat only when it strikes a body of matter and is absorbed. In considering sensible heat, the authors will deal with the absorption of radiant energy by bodies in the room or building, including the atmosphere.

Radiant energy covers a very wide range of waves, from the extremely short waves of the *cosmic ray* to the extremely long waves of radio broadcasting. This discussion is confined to the rather narrow band of radiant energy, included between the wave lengths of 0.292 microns (μ), the limit of solar radiation in the ultra-violet to 30.0 microns, the maximum wave length measured for radiation from a body at 126 F. In this connection it is especially worthy of note that the limit of solar radiation in the infra red reaching the earth is given at a wave length of 5.3 microns.

Emission of Radiant Energy

All the bodies at any temperature above absolute zero are continually emitting radiant energy. Most of the energy radiated from bodies at low temperature, however, is at long wave lengths, and therefore lies in the infra-red portion of the spectrum. If the temperature of such a body were slowly raised, the wave lengths of the maximum energy given off would shift toward the shorter wave lengths until at about 900 F a considerable portion of this energy would be at wave lengths less than 0.77 microns and would therefore be visible to the eye, and the body would have a dull red color. If heated to a still higher temperature, the wave lengths of the maximum energy would become shorter and shorter and the color of the body would change to yellow and finally to white. At very high temperatures a considerable portion of the energy is at wave lengths shorter than the visible (below 0.38 microns) and would therefore be in the ultra-violet portion of the spectrum.

While glass is transparent to energy in the visible portion of the spectrum, it is opaque to all other wave lengths except a narrow band of the longest of the ultra-violet and shortest of the infra-red. The portion of the ultra-violet and infra-red that is transmitted to some extent is cut off by thick glass, and even comparatively thin glass cuts off a large part of it.

The amount of total energy radiated at all wave lengths by a body can be represented by the area under a curve plotted with the intensity of the radiation at different wave lengths as ordinates and the wave lengths as abscissae. The fractional part emitted between any two wave length limits can be determined by comparing the area under this portion of the curve with the total area under the curve. Figs. 1, 2, 3 and 4 were drawn in this manner using the value for intensity at different wave lengths given in the International Critical Tables Vol. V.

The work of Coblentz [3] has shown that ordinary window glass 2 mm thick is opaque to all radiant energy having wave lengths shorter than about 0.3 micron and longer than 4.5 microns. Glass then is transparent to some extent to the ultra-violet between 0.3 and 0.37 microns, the visible portion of the spectrum from 0.37 to 0.78 microns, and the infra-red from 0.78 to 4.5 microns. The amount transmitted, however, is not uniform for all these wave lengths, but is selective for different wave lengths and never reaches 100 per cent in any case.

Transmission of Radiant Energy at Various Wave Lengths

Fig. 1 is derived from data covering the radiation from a body at 126 F, and from a study of this curve it is evident that the percentage of energy radiated from a body of that temperature which lies between 0.3 microns and 4.5 microns is very small, being in the order of 0.556 per cent of the total energy available. Considering only that portion from 0.3 to 4.5 microns wave length and particularly the heavily shaded area under the curve (Fig. 1), it becomes evident that glass actually transmits only a very small portion of the conceivably transmissible energy, namely, about 9.46 per cent. This percentage becomes insignificantly small when referred to the total radiated energy, and amounts to only 0.0523 per cent of that total.

As already stated, the wave lengths radiated from a body become progressively shorter as the temperature increases, and it is evident from Fig. 2 that radiations from a body at 450 F which conceivably might be transmitted by glass are considerably greater than in the case of a body at 126 F. In this case the energy which is actually transmitted amounts to only 1.19 per cent of the total energy radiated. Figs. 3 and 4 represent the total energy available and transmissible by glass at the temperatures of a tungsten filament in a projection lamp, and of the sun itself, respectively.

[1] Technical Sales Engineer, Pittsburgh Plate Glass Company.
[2] Research Laboratories, Pittsburgh Plate Glass Company.
Presented at the 38th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January, 1932.

[3] U. S. Bureau of Standards Papers, Vol. 14, 1918-19.

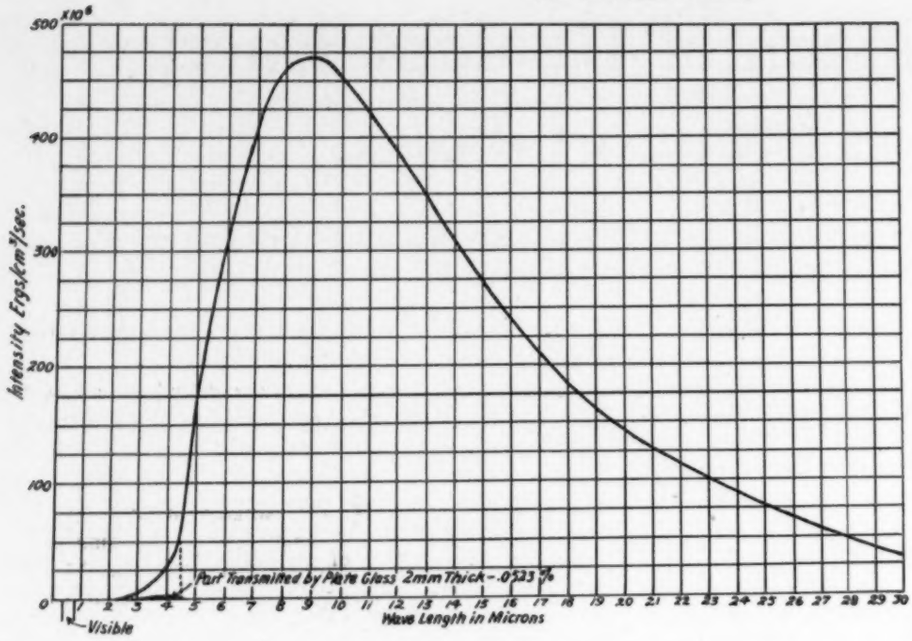


FIG. 1—CURVE OF INTENSITY OF RADIATION AT DIFFERENT WAVE LENGTHS FROM A BLACK BODY AT 126 F

This series of curves illustrates the fact that glass does not transmit radiant energy of all wave lengths proportionately, but only that very narrow band of wave lengths in the immediate vicinity of one micron on the chart of relative wave lengths (Fig. 5); and even within these limits, the transmission varies from zero to a maximum and back to zero again, in somewhat irregular order. In no case does glass transmit 100 per cent of the incident energy, since approximately 4½ per cent of the incident visible light is reflected from each surface, and in the invisible portion of the spectrum the reflectance runs as high as 19.2 per cent in the infra-red region.

It is this inability of glass to transmit appreciable amounts of long wave radiation which makes possible the growing of plants in green houses and hot frames. The short waves of solar infra-red radiation pass freely through the glass, are absorbed within the soil and other content, and again irradiated in the form of low temperature rays of long wave length, which are trapped within the hot house.

Verification of Results by Test

To test the validity of the foregoing calculated values, an apparatus was set up as shown in Fig. 6. An inverted electrically heated radiating surface *A* was kept at a definite temperature (450 F) by the automatic control *B*. This temperature was chosen because of the availability of instruments to measure radiant energy emitted at this temperature. The hot inverted surface *A* was protected from drafts by a guard *C*. *D* was a fixed-focus, total-radiation pyrometer specially designed to measure low temperatures. It was trained on the portion of the hot surface inside the guard and was connected to the automatic recorder *R* which was calibrated in millivolts. Provision was made for inserting glass

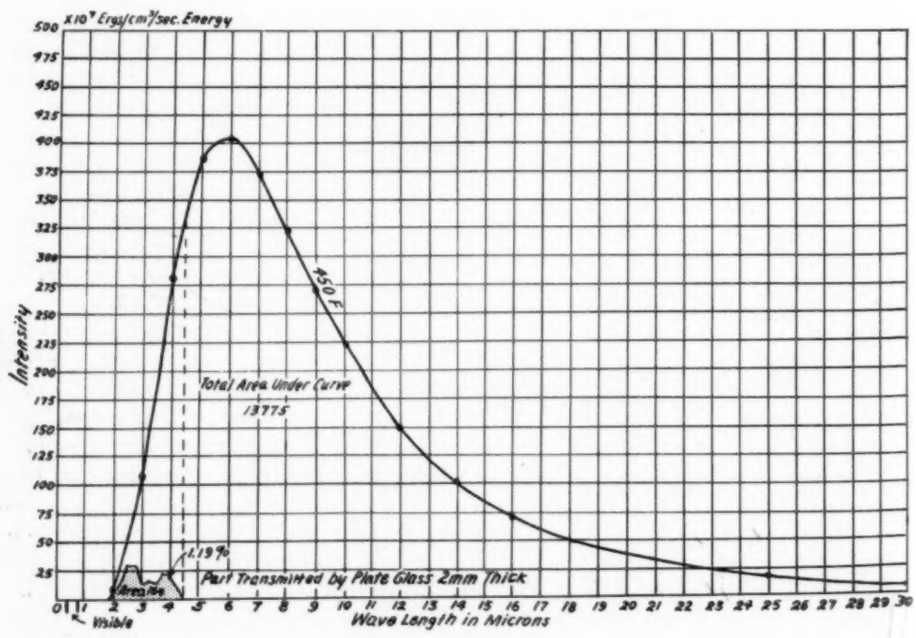


FIG. 2—CURVE OF INTENSITY OF RADIATION AT DIFFERENT WAVE LENGTHS FROM A BLACK BODY AT 450 F

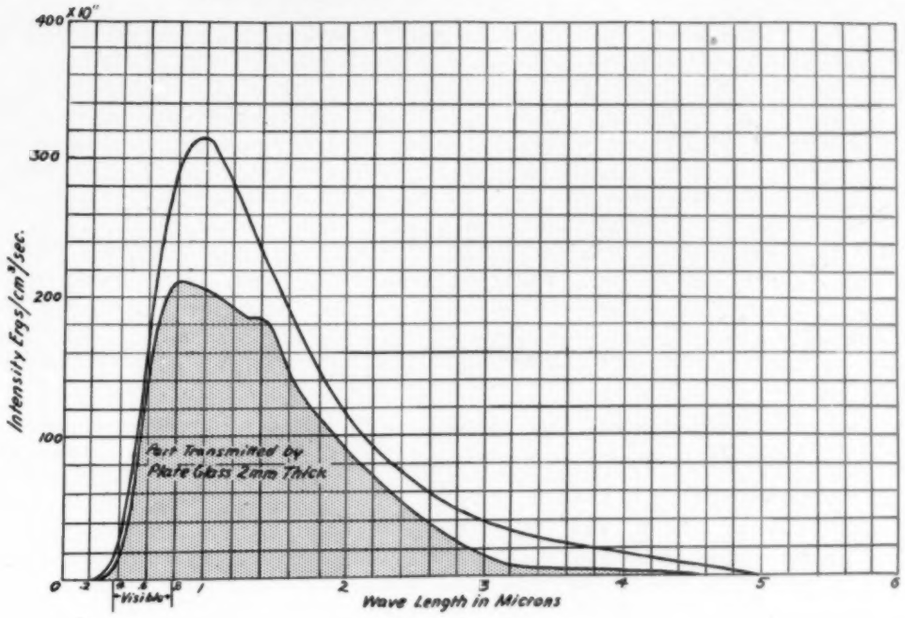


FIG. 3—CURVE OF INTENSITY OF RADIATION AT DIFFERENT WAVE LENGTHS FROM A BLACK BODY AT 3000 K* (TUNGSTEN FILAMENT TEMPERATURE)

*Degrees Kelvin.

plates in the holder *G* between the radiating surface and the pyrometer. The inverted position was used so that convection currents could not heat the glass and so that it would be affected only by the energy radiated from the hot surface.

After the radiating surface had reached a constant temperature (450 F), the recorder was allowed to run without any glass between the source and the pyrometer until a curve had been traced on the chart. This was of course a straight line as the temperature of the radiating surface was constant and indicated a constant deflection of 2.47 mv (millivolts). A single plate of glass ⅛ in. thick was now inserted between the pyrometer and the radiating surface (Fig. 7) and the deflection of the recorder at once fell to zero, showing that practically no radiation was passing through the glass. This seems to check the curves of Figs. 1 and 2 as derived from the Coblentz data. The deflection was actually less than 0.05 mv and was too small to be indicated by the recorder. As the radiant energy was absorbed by the glass, the top surface began to rise in temperature and this heat was of course conducted through the glass to the bottom surface and was re-radiated there as shown by a gradual rise in the curve traced by the recorder.

The pyrometer was now receiving the radiant energy from the glass instead of the hot surface. When the glass had reached an equilibrium in temperature, the recorder again traced a straight line. The constant deflection, however, was much less than before due to part of the original energy incident on the glass being reflected, part being radiated from the top surface and part of the energy of the hot glass being lost through conduction to the air touching the bottom and the top surface. The constant deflection of the pyrometer was 0.52 mv. Thus about 21 per cent of the original energy

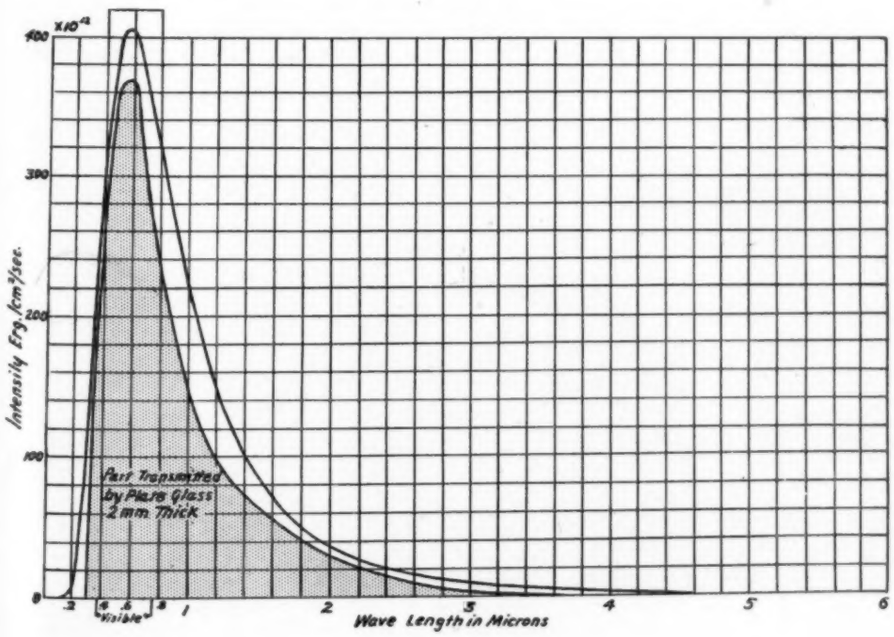


FIG. 4—CURVE OF INTENSITY OF RADIATION AT DIFFERENT WAVE LENGTHS FROM A BLACK BODY AT 5000 K* (SUN TEMPERATURE)

*Degrees Kelvin.

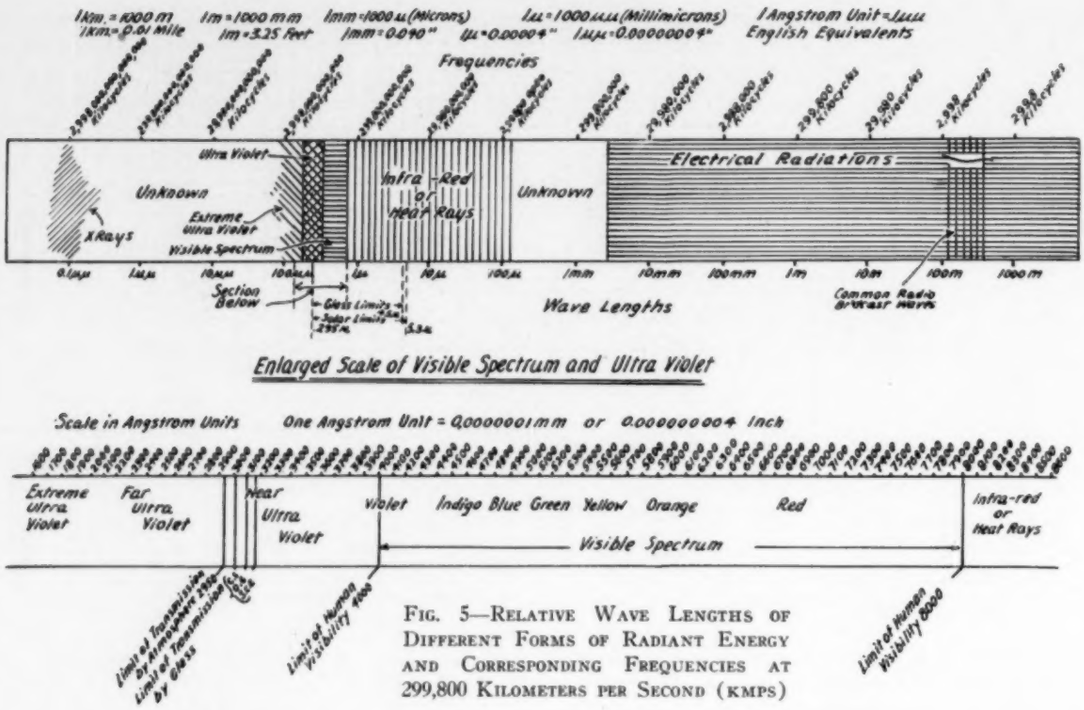


FIG. 5—RELATIVE WAVE LENGTHS OF DIFFERENT FORMS OF RADIANT ENERGY AND CORRESPONDING FREQUENCIES AT 299,800 KILOMETERS PER SECOND (KMPS)

incident on the glass was transmitted or radiated from the bottom surface. This is shown on Curve V of Fig. 8.

Interpretation of Results

If it is assumed that the temperature of both surfaces of the thin plate of glass were practically the same, a rough calculation of the total losses can be made. As already stated, 21 per cent was re-radiated from the bottom side, including the 1 per cent transmitted, and since a hot body radiates uniformly in all directions, 20 per cent would also be radiated back from the top surface. It will be assumed that 5 per cent was reflected back by the top surface and 1 per cent was transmitted as radiant energy by the glass. Of the total, 47 per cent has been accounted for and hence, 53 per cent was lost by conduction to the air from the bottom and top surfaces. If it is assumed that this conduction to the air was uniform on both sides, the loss by conduction from the bottom surface amounted to 26.5 per cent which, added to the 21 per cent lost by radiation and transmission, amounts to a total loss of 47.5 per cent of the original energy from the bottom surface of the glass.

Curve VI of Fig. 8 shows the results of an exactly similar experiment, except that two plates of ⅛-in. glass were separated by a ⅜-in. air space. This experiment indicated that about 10 per cent of the energy incident on the upper glass surface reaches the pyrometer by radiation from the bottom surface of the lower plate together with that passing through by direct transmission. This is a greater loss than would be expected but it can be accounted for by noting that the bottom plate in this setup is receiving all of its energy from radiation of the heated top plate as it can be assumed that the conduction through the air space is negligible. In the case of the single plate, the temperature of the glass finally reached 220 F but with the two plates, the top plate reached a

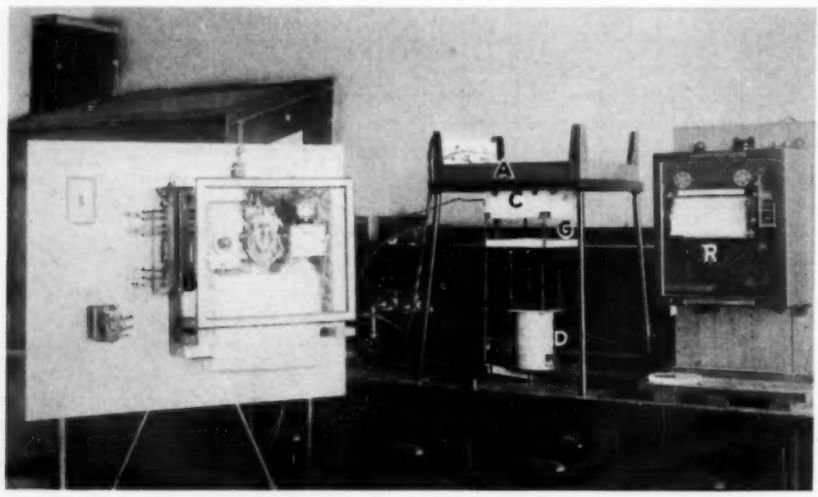


FIG. 6—VIEW OF TEST APPARATUS

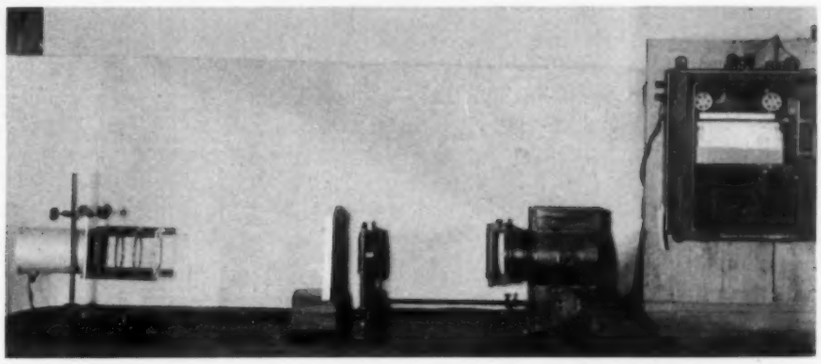


FIG. 7—VIEW OF APPARATUS, TUNGSTEN FILAMENT USED AS A SOURCE OF ENERGY

temperature of 275 F. This was because the heat loss by conduction from the bottom surface of the upper plate was stopped by the air space and also some of the energy radiated by the bottom surface was radiated and reflected back by the lower plate. Because of this increase in temperature, considerably more energy than the original 21 per cent was radiated from the bottom surface of the top plate. Part of this energy is reflected back by the lower plate as mentioned before and the remainder is absorbed by the lower plate, then part re-radiated back to the upper plate and the remainder lost from the bottom surface of the lower plate by radiation and conduction to the air. The lower plate reached a final temperature of 140 F.

As mentioned before, it was found that the amount of energy lost from the lower plate by radiation was 10 per cent of the total energy incident on the upper plate. Assuming that the amount of energy lost by conduction in each case was proportional to the absolute temperature difference between the air (80 F) and the glass, it was calculated that 11.5 per cent of the original energy was lost from the bottom surface of the lower plate by conduction to the air. Thus a total of 21.5 per cent of the total energy incident on the upper plate was lost from the lower plate to the outside. Since the total loss with a single plate was 47.5 per cent, the introduction of a second plate and an air space reduced this loss to less than half.

To show that this conservation was not due merely to the added thickness of glass, the two plates were replaced with a single plate of ½-in. plate glass. Although the thickness was twice as great as the combined thicknesses of the two plates, 16 per cent of the total incident energy was radiated from the bottom surface (Fig. 9) compared to 10 per cent in the case of two thin plates separated by an air space.

The foregoing calculations are not of course absolutely accurate but are intended to show only the approximate conditions. It should be borne in mind that these temperatures are considerably higher than those employed in heating human habitations, and therefore, that the proportion of transmissible wave lengths is considerably in excess of those encountered in household temperature conditions.

A significant feature of all of these curves is the fact that in every case, as soon as the glass was placed be-

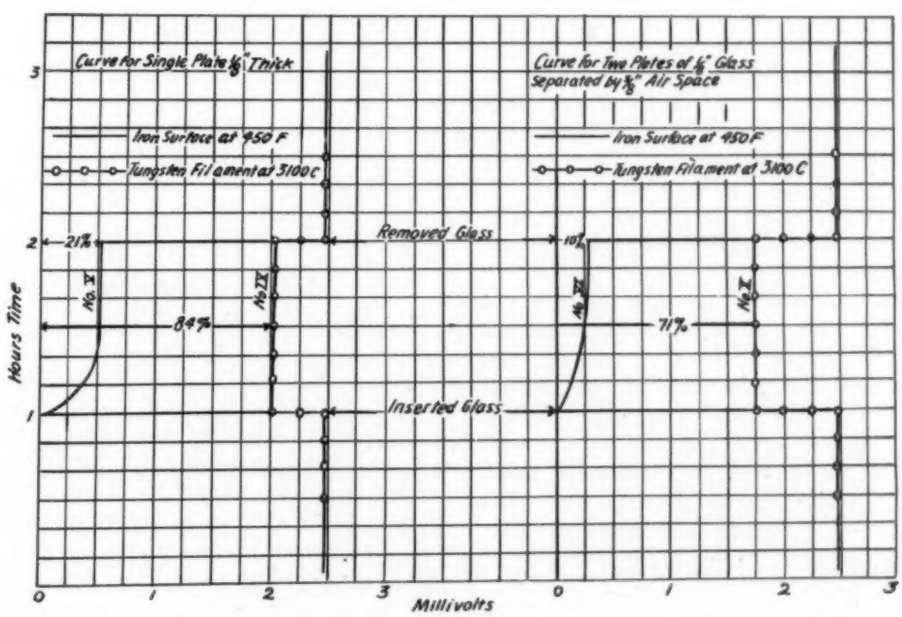


FIG. 8—CURVES DRAWN BY RECORDING PYROMETER PEN FOR SINGLE PLATE GLASS ⅛-IN. THICK AND TWO PLATES OF ⅛-IN. GLASS SEPARATED BY A ⅜-IN. AIR SPACE

tween the pyrometer and the source of energy, the deflection fell to zero. This clearly indicates that glass is opaque to radiations from bodies at low temperature and that radiant energy was again received by the pyrometer only after the glass itself was heated.

Due to the nature of the set-up, the only energy transfer between the hot surface and the glass plates was by direct radiation. If the set-up had been modified so that the energy could also have been transferred from the hot surface to the glass by convection currents, the losses through a single plate of glass would have been far greater.

Referring again to Fig. 8, curves IX and X show the actual results obtained when light at an approximate temperature of 3,000 C was used as the source of energy. The apparatus used is shown in Fig. 7. In this instance, it will be noted that the screening value of an increasing number of plates is more nearly proportional to the number of plates involved. There is very little of the energy of visible light which is not transmissible by glass and in general, only that part is screened out which is reflected from the surfaces, since very little energy of these shorter wave lengths is absorbed by the glass.

The transfer of sensible heat through glass is, of course, subject to the same laws as in the case of any other material. It will depend upon conductances, film interchange as between the glass and the atmosphere, and vice versa, and the velocity of air flow over the glass surfaces. By calculation, based upon accepted determinations, it may be shown that three plates of ⅛-in. glass spaced approximately ½ in. apart are equivalent in insulating value to a 12-in. brick wall.[4] If this air space is reduced, the insulating value approaches that of an 8-in. brick wall.

[4] See Chapter 3, A. S. H. V. E. Guide 1932. The coefficient of transmission of a 12-in. brick wall (no interior or exterior finish) is given as 0.295 Btu per hour per square foot per degree Fahrenheit difference in temperature whereas that for triple glass is given as 0.281.

The energy from the sun could easily enter through the glass to be absorbed and be retained within the room, but could not enter through a brick wall. That this heating by transmitted sun rays is considerable, is illustrated by a closed automobile that has been exposed for some time to the bright sunshine. Even when the outside temperature is zero, the inside of the car will be comfortably warm with no other means of heating the interior except by the transmission of the sun's rays through the glass.

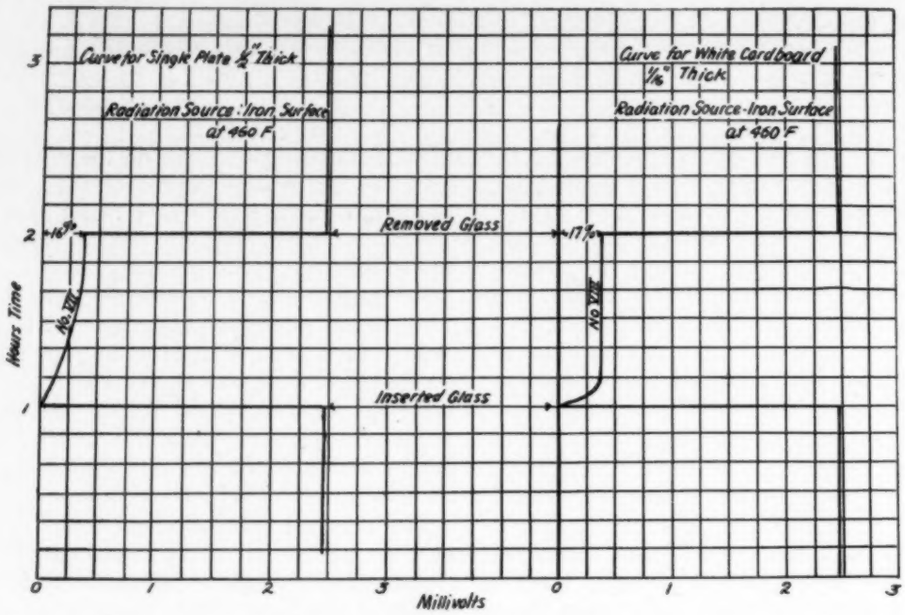

Curve for Single Plate ½" Thick
Radiation Source: Iron Surface at 460 F
Curve for White Cardboard 1/16" Thick
Radiation Source-Iron Surface at 460 F
Removed Glass
Inserted Glass
Hours Time
Millivolts

FIG. 9—CURVES DRAWN BY RECORDING PYROMETER PEN FOR SINGLE PLATE GLASS ½-IN. THICK AND WHITE CARDBOARD 1/16 IN. THICK

Again in considering the transfer of sensible heat through glass, it is of course obvious that there is a maximum spacing between plates which will be most effective. The maximum spacing for good results seems to be about ½ in. and the optimum spacing seems to be between ⅛ in. and ⅜ in. Preliminary investigations are being conducted along this line, and the insulating value of double glass with varying interglass spaces is being studied. It is interesting to note that these preliminary investigations indicate that there is very little difference between spacings of ⅜ in., ½ in. or ⅝ in., since the gradient shown under the same conditions seems to be 51 deg, 53 deg, and 50 deg respectively.

Conclusions

There is practically no loss of energy from a building by direct transmission of radiant energy through the windows. Some energy is lost by the glass absorbing the radiant energy and thereby being heated and then radiated to the outside. This loss, however, is believed to be insignificant compared to the losses due to convection currents and conduction. Two plates of glass separated by an air space cut off the greater part of the radiation loss and reduce the conduction loss to a comparatively low figure.

It seems evident that the increasing use of glazed areas in buildings need not mean increased heat losses if multiple panes of glass, separated by air spaces, are used.

PITTSBURGH EXPERIMENT STATION OF THE U. S. BUREAU OF MINES WHERE THE RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS IS LOCATED

Report of the Committee on Research for 1931

By C. V. Haynes, *Chairman*, and F. C. Houghten, *Director of Research Laboratory*

THE Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS gave consideration to 12 research projects during the year. In connection with each of these ventures, a sub-committee of the Committee on Research, known as a Technical Advisory Committee, was appointed for the purpose of studying the needs of the problem, and to cooperate with the Director of the Laboratory in plans for research to be carried on in the Laboratory at Pittsburgh or in the cooperating institutions. Each of these Technical Advisory Committees was made up of authorities on the particular subject. During the past year this organization brought to the Society's research activity the combined experience and ability of 78 experts.

The Committee on Research was very fortunate in securing the services of Prof. A. C. Willard as Technical Adviser to the Committee on Research. This addition to the organization is of great value, as it brings the broad experience and ability of this noted engineer and educator to aid in the work. Other slight changes in organization were made in order to better correlate the research activities with the other activities of the Society.

Five meetings of the Committee on Research were held during the year: two during the Annual Meeting in Pittsburgh, one in Chicago in the spring, one at Swampscott during the Summer Meeting of the Society, and the final meeting at Buffalo in November. At these meetings the welfare of the Laboratory was discussed including its finances, the development of a research program, and the approval of cooperative agreements for research.

The funds for the Laboratory work were definitely allocated in the budget and as in the past were derived from dues of Society members, interest on reserve funds, THE GUIDE, and contributions from the industry, which were most generous considering present financial conditions. This made a total commensurate with previous years. New contracts and greater contributions to institutions cooperating with the Laboratory made it necessary to slightly revise the budgeted expenditures at Pittsburgh.

Presented at the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1932.

Income and Expenses—Committee on Research

Income	*Budget* 1931	*Actual* 1931
A. S. H. V. E. Dues and Guide	$19,000.00	$22,519.85
Pledges and Contracts		
Manufacturers and Associations	13,110.00	10,283.00
Other Contributions	15,000.00	
Interest on Bank Balances	200.00	71.21
Interest on Securities	200.00	135.00
Interest from Society Endowment	1,000.00	1,025.00
	$48,510.00	$34,034.06
Expenses		
Committee on Research		
Travel—Committee and Lab. Personnel	$2,800.00	$1,688.21
Salary Clerical—Society Headquarters	800.00	800.00
Publicity and Promotion Work	2,000.00	1,207.34
Assn. for Correlating Thermal Research	500.00	500.00
	$6,100.00	$4,195.55
A. S. H. V. E. Laboratory (Pittsburgh)		
Salaries	$19,250.00	$18,517.63
Laboratory Supplies and Equipment	2,000.00	1,131.36
Office Supplies and Expenses	800.00	688.77
Meetings	200.00	93.75
Contingent	400.00	
Contracts at Cooperating Institutions	14,950.00	14,448.60
	$43,700.00	$39,075.60

1931 Contributors to Research Fund

Manufacturers

Aerofin Corporation, Newark, N. J.
American Blower Corp., Detroit, Mich.
Barnes and Jones, Boston, Mass.
Buffalo Forge Co., Buffalo, N. Y.
Carrier Engineering Corp., Newark, N. J.
Crane Company, Chicago, Ill.
Hoffman Specialty Co., Waterbury, Conn.
Ilg Electric Ventilating Co., Chicago, Ill.
Nash Engineering Co., South Norwalk, Conn.

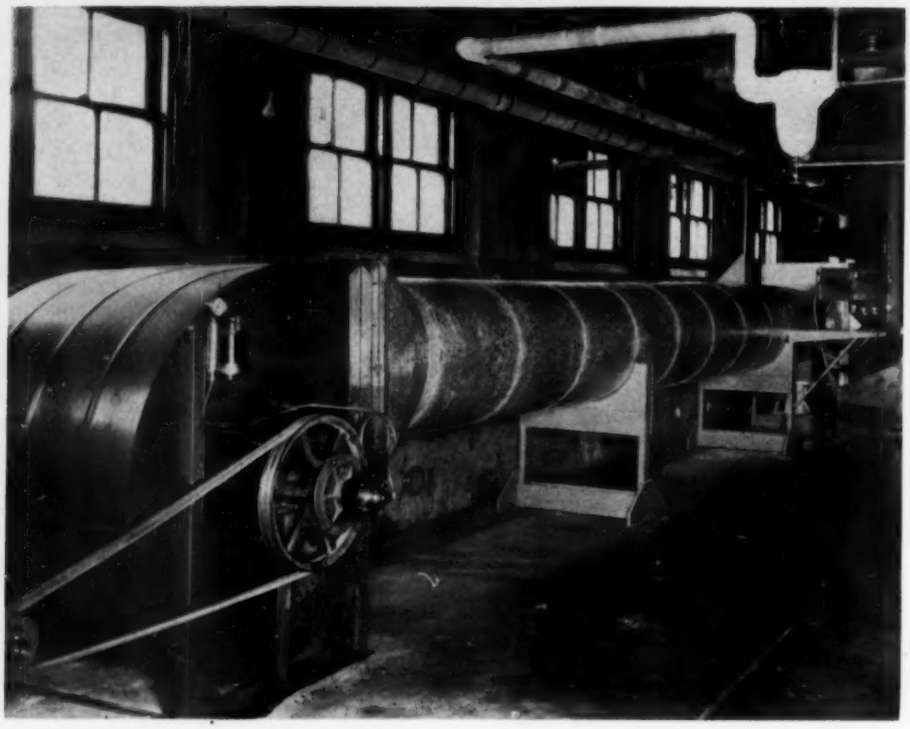


FIG. 1—APPARATUS AT THE UNIVERSITY OF MINNESOTA USED FOR STUDYING SURFACE COEFFICIENTS AS AFFECTED BY DIRECTION OF WIND

Powers Regulator Co., Chicago, Ill.
Trane Company, La Crosse, Wis.
United Engrs. & Constructors, Philadelphia, Pa.
Warren Webster and Co., Camden, N. J.
Wolverine Tube Co., Detroit, Mich.

Associations

Associated Copper Tubing Mfrs.
American Oil Burner Assn.
Ventilating Contractors Employers Assn. of Chicago.
National Lumber Manufacturers Assn.

Cooperative Research

Cooperative research programs have been arranged for by contract with the following institutions where the projects listed are being investigated:

University of Minnesota: Heat transmission through built-up walls, determination of surface coefficients, conductivity of insulating materials, and the thermal properties of different species of wood. The latter study is carried on through the cooperation of the *National Lumber Manufacturers Association* with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University of Minnesota.

University of Illinois: The study of the performance characteristics of radiators.

Harvard School of Public Health: The ionization of air in its relation to health.

University of Wisconsin: The airation of buildings due to infiltration and other air exchanges.

Yale University: Development of standard test methods for oil burning devices in steam and hot water heating boilers.

Carnegie Institute of Technology: Study of steam, condensate and air flow in low pressure steam heating systems.

Agricultural and Mechanical College of Texas: Frictional resistance to the flow of water in hot water heating systems.

Armour Institute of Technology: Measurement of air flow through registers and grilles, including air delivery from unit ventilators.

Research Subjects Considered

Of the 12 research subjects given consideration by the Committee on Research and its Technical Advisory Committees, 7 were actively investigated at the Laboratory in Pittsburgh or in one of the cooperating institutions. From these studies, 16 papers were prepared for publication and presentation at either the 1931 Summer or the 1932 Annual Meeting of the Society. All of these papers served to add to the fundamental technical knowledge of the heating and ventilation branch of the engineering profession and industry.

The following 8 technical papers resulting from research at the Laboratory and in the cooperating Universities are being presented at the Annual Meeting:

Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (*Heating, Piping and Air Conditioning*, September, 1931).

Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods, by C. P. Yaglou, L. Claribel Benjamin, and Sarah P. Choate (*Heating, Piping and Air Conditioning*, October, 1931).

Heat Emission from Iron and Copper Pipe, by F. C. Houghten and Carl Gutberlet (*Heating, Piping and Air Conditioning*, January, 1932).

Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and W. A. Eckley (*Heating, Piping and Air Conditioning*, October, 1931).

Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott.

Supplementary Friction Heads in One-Inch Cast Iron Tees, by F. E. Giesecke and W. H. Badgett (*Heating, Piping and Air Conditioning*, January, 1932).

A Study of the Intermittent Operation of Oil Burners, by L. E. Seeley.

Field Studies of Office Building Cooling, by S. S. Sanford, E. P. Wells and J. H. Walker, which is to be presented at the Annual Meeting, is the result of research in which the Research Laboratory cooperated to a lesser extent.

A paper, Heat and Moisture Dissipation from Children in Relation to School Ventilation, by F. C. Houghten, W. W. Teague, and W. E. Miller, was presented at the Annual Meeting of the *American Public Health Association* in Montreal in September, 1931.

The following 12 subjects were given consideration during the past year by the Committee on Research and the Technical Advisory Committees. A brief review of the work of the Technical Advisory Committees and the Laboratory or cooperating universities on these projects is given together with the members of the Society serving on these sub-committees.

1. HEAT TRANSMISSION

(Heat Received and Emitted by Buildings in Relation to Living Comfort)

Technical Advisory Committee

L. A. Harding, *Chairman*
R. E. Backstrom
F. B. Rowley
A. E. Stacey
J. H. Walker

During the past year the Technical Advisory Committee on Heat Transmission gave consideration to several research problems. Some of these had been approved previously for study at the University of Minnesota in cooperation with the Research Laboratory and at the Laboratory in Pittsburgh. The committee also gave consideration to additional problems and submitted a recommended program for the consideration of the Committee on Research.

a. A STUDY OF THE THERMAL CONDUCTIVITIES FOR VARIOUS TYPES OF BUILT-UP WALL CONSTRUCTION BY THE HOT-BOX METHOD.

Work carried on under the direction of Professor Rowley at the University of Minnesota in cooperation with the Research Laboratory.

This work is a continuation of that carried on last year. The program was extended to include several varieties of hollow tile, hollow concrete block, monolithic concrete, cinder concrete block walls, and several additional types of frame wall construction.

About 40 walls have been built up and tested since the last report was published. A report of these tests is now being prepared for publication during the next few months.

A series of tests is now in progress to determine the effect of changing air temperature on the thermal conductivity of uninsulated and insulated walls. This work will parallel and correlate with the work now being carried on at the Pittsburgh Laboratory. Several tests have been made to determine the effect of the type of surfaces bounding air spaces on the thermal conductivity of the air space.

b. STUDY ON SURFACE COEFFICIENTS WITH WIND NOT PARALLEL TO THE WALL.

Work carried on under the direction of Professor Rowley at the University of Minnesota in cooperation with the Research Laboratory.

The study of surface heat transfer coefficients for still and moving air was extended during the early part of this year to determine the effect of the angle of wind impingement upon surface. A paper covering this part of the work has been prepared and will be presented at the Annual Meeting of the Society, January 1932.

c. THERMAL CONDUCTIVITY OF INSULATING MATERIALS

Work carried on under the direction of Professor Rowley at the University of Minnesota in cooperation with the Research Laboratory.

A series of tests have been made to determine the thermal conductivity of various types of insulating materials now on the market. This work has been confined almost entirely to the board or rigid forms of materials, although it is now being extended to include the quilt or flexible materials and fills. Samples of the different materials have been collected from various parts of the United States in order to get representative materials. It is hoped that from this series of tests average values may be obtained which may be used in the A. S. H. V. E. GUIDE.

c. CONDUCTIVITY OF VARIOUS SPECIES OF LUMBER.

Work carried on under the direction of Professor Rowley at the University of Minnesota in cooperation with the Research Laboratory.

A series of tests started late in 1930 to determine the thermal properties of various species of lumber has been in progress throughout the year. This work is a cooperative project between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory, the *National Lumber Manufacturers' Association* and the University of Minnesota. The work has been financed by the *National Lumber Manufacturers' Association.*

The series of tests will cover about thirty species of wood and will include the effect of moisture and density on the conductivity for the various species. Some difficulty was encountered in getting consistent results on several of the species. This required a more extensive program than was first anticipated.

d. VARIATION IN CONDUCTIVITY OF CONCRETE.

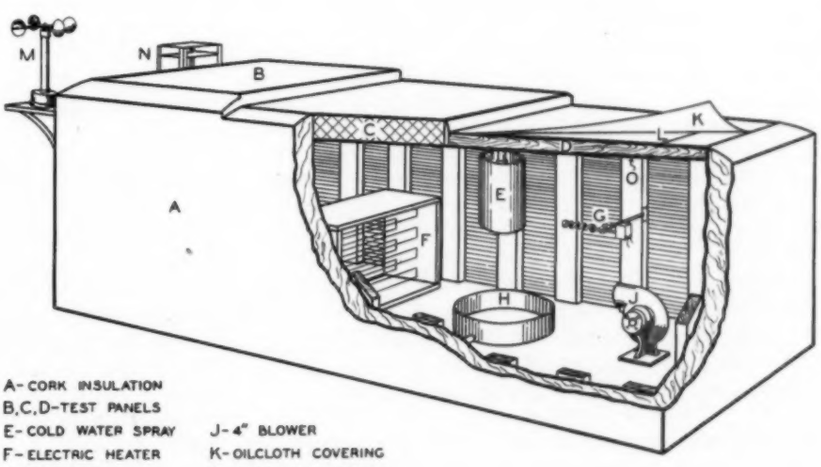
Work carried on at the Research Laboratory in Pittsburgh.

This project has been under investigation for several years, and has had as its object the determination of the variation in conductivity of concrete with aging. Two samples were tested periodically, showing a decrease in conductivity with time. The tests during the past year showed very little additional change. However, the samples are being kept with a view of testing them at longer intervals in the future. A paper resulting from this work will be presented at the Annual Meeting of the Society, January 1932.

e. EFFECT OF SOLAR RADIATION AND HEAT CAPACITY OF A STRUCTURE ON HEAT FLOW INTO A BUILDING.

Work carried on at the Research Laboratory in Pittsburgh.

This work was first undertaken in 1930, when a study was made of three roof panels made up of laminations of iron and



A-CORK INSULATION
B,C,D-TEST PANELS
E-COLD WATER SPRAY
F-ELECTRIC HEATER
G-THERMOSTAT
H-SPRAY RECEIVER
J-4" BLOWER
K-OILCLOTH COVERING
L-HEAT FLOW METER
M-CUP ANEMOMETER
N-SUN SHIELD FOR AIR COUPLE

FIG. 2—AIR-CONDITIONED CHAMBER AT THE A. S. H. V. E. RESEARCH LABORATORY FOR ANALYZING THE HEAT FLOW THROUGH ROOF PANELS

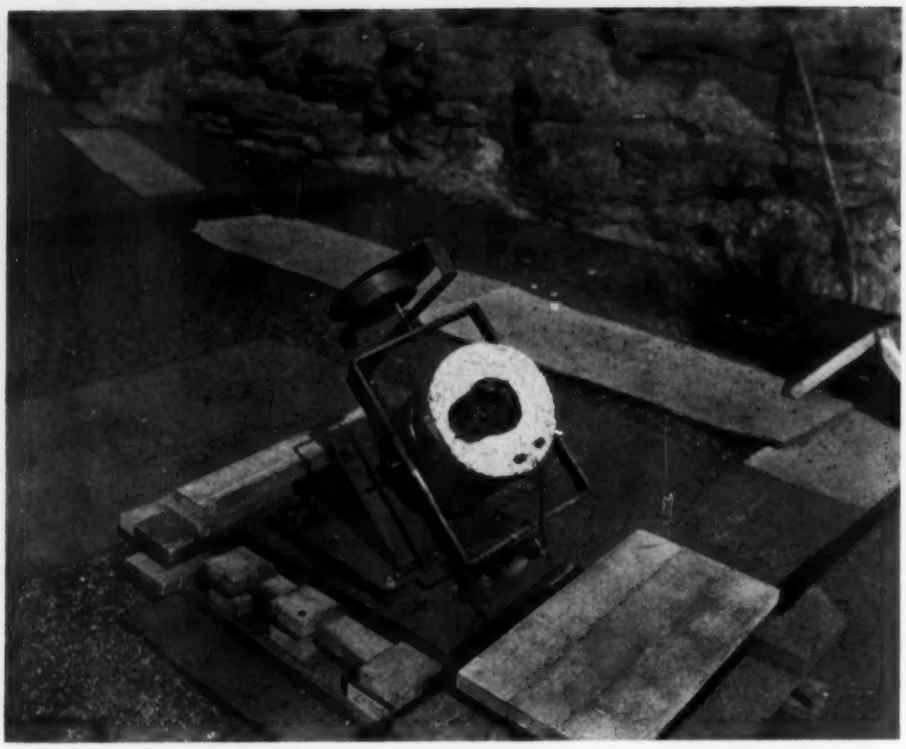


FIG. 3 — PYRHELIOMETER AT THE A. S. H. V. E. RESEARCH LABORATORY FOR AUTOMATICALLY RECORDING THE INTENSITY OF SOLAR RADIATION. THIS INSTRUMENT WAS USED IN OBTAINING DATA ON HEAT TRANSMISSION AS INFLUENCED BY HEAT CAPACITY AND SOLAR RADIATION

cork. Early during the present year the Technical Advisory Committee gave consideration to an extension of this program which was approved by the Committee on Research. The extended program included a study of heat flow through the following roof panels during the hot summer months:

1. 2-in. tongue and groove yellow pine,
2. 4-in. gypsum cast in place,
3. 6-in. concrete
4. 3-in. concrete with 1-in. corkboard insulation with concrete up.
5. 3-in. concrete with 1-in. corkboard insulation with cork up.

This extended program was completed during the summer, and a paper will be presented at the Annual Meeting of the Society, January 1932. The Committee on Research at its meeting held in Buffalo in November, 1931 approved the continuation of this project.

f. WIND VELOCITY GRADIENTS NEAR A WALL.

Work to be carried on at the Laboratory in Pittsburgh.

A year ago the Laboratory published a paper resulting from work in Pittsburgh showing wind velocity gradients near a wall surface for conditions where the direction of the wind was parallel to the wall surface. These data were obtained by mounting a wall on the side of a truck and carrying it forward. Further consideration has been given by the Technical Advisory Committee to the possibility of verifying these findings for a condition of natural wind. The Technical Advisory Committee approved continuing this study, and recommended the same to the Committee on Research. At a meeting of the Committee on Research, held in Buffalo in November, this project was approved for study at the Laboratory in Pittsburgh during the latter part of the present winter.

g. THE COOLING DEMAND IN AN OFFICE BUILDING DURING THE SUMMER.

Work carried on by J. H. Walker in cooperation with the Research Laboratory

Early during the present year the Committee on Research approved cooperation between the Research Laboratory and J. H. Walker in accordance with which the Laboratory loaned available instruments for and acted in an advisory capacity in a study of cooling demand in an air-conditioned office building in Detroit. The study included an analysis of the sources of heat gain by the building during the hot summer months.

2. PIPE SIZES

Pipe and Tubing (Sizes) Carrying Low Pressure Steam and Hot Water

Technical Advisory Committee

S. R. Lewis, *Chairman*
J. C. Fitts
F. E. Giesecke
H. M. Hart
C. A. Hill
A. P. Kratz
W. K. Simpson

The study of the capacity of pipe for carrying steam, water and air in low pressure steam heating systems has been carried on by the Laboratory in Pittsburgh for the past several years. As a result of these studies, the steam heating pipe size section of THE GUIDE has been revised from year to year. The end of these extensive investigations is now in sight and the problem will probably be concluded during the coming year. All values in THE GUIDE pipe size tables will then be based upon sound experimental data.

During the year the Laboratory in Pittsburgh made an analysis of THE GUIDE tables. As a result of this analysis, the Technical Advisory Committee at a meeting during the Semi-Annual meeting of the Society agreed that the following information was still desirable before making a final revision of THE GUIDE tables:

First, air and condensation loads on the return side of gravity return and vacuum return steam heating systems both during and after the heating-up period; *Second,* determination of the capacity of 1½-in. and 2-in. dry return mains, risers and wet return mains in gravity and vacuum pump systems. These two projects have been approved for immediate study by the Laboratory in Pittsburgh.

During the past year, the Laboratory in cooperation with the *Copper Tubing Manufacturers' Association* has made a study

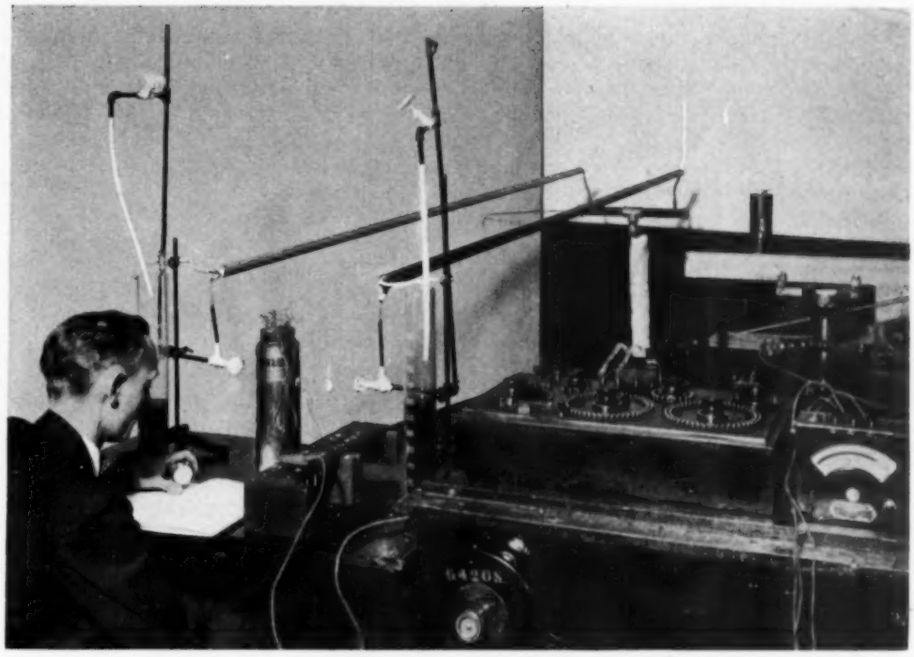


FIG. 4—SET-UP AT A. S. H. V. E. RESEARCH LABORATORY FOR DETERMINING HEAT EMISSION FROM HORIZONTAL PIPE. THIS APPARATUS WAS USED IN OBTAINING DATA ON HEAT EMISSION FROM IRON AND COPPER PIPE

of the performance of the copper tubing in steam and hot water heating systems. The studies relating to steam heating systems were carried on in Pittsburgh, and those relating to hot water systems were carried on under the direction of Professor Giesecke at the Agricultural and Mechanical College of Texas in cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

The study of performance characteristics of copper tubing in steam heating systems included: *First,* a study of the heat emission from the surface of bare and insulated copper tubing and iron pipe; *Second,* a study of the relative capacity of copper tubing for carrying steam and condensate in vertical and horizontal straight runs of pipe and in radiator supply branches. These studies on copper tubing have been completed, and a paper, Heat Emission from Iron and Copper Pipe, will be presented at the Annual Meeting, January 1932.

The study of frictional resistance to flow of water in hot water heating systems was continued by Professor Giesecke of the Agricultural and Mechanical College of Texas in cooperation with the Research Laboratory. The present investigation includes a study of the frictional resistance in orifices and standard iron tees and frictional resistance in copper tubes and fittings. A paper will be presented at the Annual Meeting of the Society, January 1932.

3. AIR CLEANING

Atmospheric Dust and Air Cleaning Devices (Including Dust and Smoke)

Technical Advisory Committee

H. C. Murphy, *Chairman*
Albert Buenger
Philip Drinker
Dr. E. V. Hill
H. B. Meller
Dr. S. W. Wynne

Consideration was given to the problem by the Technical Advisory Committee and plans for the future are being outlined.

Early during the present year a study was carried on by Dean Langsdorf of Washington University, St. Louis, in cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The object of this study was to develop a more sensitive and more satisfactory means of determining the dustiness of air. The method was rather completely developed by Dean Langsdorf, but the instrument is not yet completely calibrated and tested out for practical use.

4. AIR FLOW

Air Flow Through Registers and Grilles

Technical Advisory Committee

John Howatt, *Chairman*
J. J. Aeberly
C. A. Booth
L. E. Davies
D. E. French
J. J. Haines

The study of methods of measuring air flow through registers and grilles is being carried on by Professor Davies at Armour Institute of Technology in cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *Ventilating Contractors Employers Association* of Chicago. The development of the work to date may be divided into the following four phases:

1. Early work using grilles of simple design on supply led to the use of the formula

$$Q = \frac{CV\,(A + a)}{2} \qquad (1)$$

in which the coefficient K averaged 0.80, was found value 1.0 that it could usually be omitted from the formula. For more accurate results a table of coefficients varying with the velocity was proposed.

2. The second stage of the research was devoted to exhaust grilles. It was found that Formula 1, while appreciably more accurate than formulae formerly in common use, did not give as consistently accurate results as it did with supply grilles. A second formula,

$$Q = KVA \qquad (2)$$

in which the coefficient K averaged 0.80, was found to be somewhat more accurate, and therefore more desirable because it did not require the measurement of free area of the grille. Again a table of coefficients varying with velocity was proposed for use where greater accuracy was desired.

3. The third stage consisted of research on grilles of more ornate designs, ornamental plaster, ornamental iron and thick cast iron. Formula 2 was found to give quite accurate results in all cases, while Formula 1 was subject

to considerable error in some cases. The work on exhaust grilles may be considered as completed. On supply, Formula 1 was found to be accurate in the majority of cases. In a few cases, however, the results were not satisfactory and it is these cases that require further study.

4. As there are several types of anemometers in use, and as the major part of this research was conducted with one type of instrument it was necessary to make a comparison of the different kinds. Considerable time was devoted to this phase of the problem, using five of the types in most common use. The results obtained with the different instruments were in very close agreement.

FIG. 5—APPARATUS AT THE AGRICULTURAL AND MECHANICAL COLLEGE OF TEXAS FOR MEASURING FRICTIONAL RESISTANCE TO THE FLOW OF WATER IN TEES

The investigation will be continued at Armour Institute of Technology and will include the following three phases:

1. Ornamental and other special types of grilles. Investigations last year showed that on supply grilles when the designs were characterized by largeness of detail (large solid parts and large openings), the results obtained with the new formula were not as accurate as would be desired, although in general more accurate than the results obtained with the formulae and methods previously used.

2. In some cases such as ornamental grilles, heater cores and cases where deflectors or heaters are located in close proximity to the grilles (unit ventilators), there is considerable doubt as to what should be considered free area.

3. In some cases, particularly in the case of long narrow grilles, the nature of the approach results in a strip of motionless or re-entering air along one edge. Where this condition extends in for several inches from the edge, the accuracy of the results will not be seriously affected, as the anemometer will come to rest at these points and thus bring down the average reading obtained by the proper amount. In the case of a long slender grille, however, a strip of quiescent air, say ¾-in. wide, will not be sufficient to cause the instrument to read lower and yet is great enough to cause considerable error if not considered in the calculations.

Considerable work has already been done on these special cases, but more should be done before a positive method can be stated for determining them. In many cases a simple modification in the procedure will be sufficient to insure reasonably accurate results. In other cases, particularly those in which it is difficult to determine the free area, some distinctly new method must be used.

Experiments have been made with a hood or tube to fit over the entrance side of the anemometer. With this device it is only necessary to know the gross area of the grille, which makes the calculations quite simple.

5. HEATING UNITS

Radiation—Direct and Indirect

Technical Advisory Committee

John Holton, *Chairman*
R. M. Conner
R. E. Daly
R. V. Frost
F. E. Giesecke
H. F. Hutzel
A. P. Kratz
J. F. McIntire
W. T. Miller
R. N. Trane

The Technical Advisory Committee on Radiation (direct and indirect) has given particular consideration during the past year to a number of problems. These include test methods to be used, specifications of test rooms, and the policy of the Laboratory regarding its entrance into the field of commercial testing of radiators.

At a meeting of the committee held during the Semi-Annual meeting of the Society, June 1931, the following six recommendations were agreed upon as reported by the chairman:

a. That for the present, no action be taken on the commercial testing of radiation, either exposed or concealed, by the Research Laboratory. The laboratory facilities can be used to better advantage along lines tending to clarify approved methods of testing all forms of radiation.

b. That codes for testing all forms of so-called *radiation,* both exposed and concealed, be developed and approved by the Society. With these as a basis, other agencies may then be in a position to do all the necessary testing, then if desired, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS may consider the proper organization and equipment to undertake this commercial testing. Until such codes are a part of the approved program of the Society, it is not in a position to render needed service to the industry.

c. That in view of the divided opinion and the many unknown factors involved in the effectiveness of the various heating mediums, it is suggested, at least for the present, that the Technical Advisory Committee on Radiation go on record as favoring a condensation method properly controlled as the measurement of the total heat delivery from such equipment. In other words, a condensation rating represents a load method and measures the amount of heat required by, and delivered to the heating equipment.

d. That the Society continue to authorize the study of the utilization or effectiveness of the heat delivered by various types of heating equipment. This involves further investigations of the so-called heating effect, and should determine

(1) Definition of heating effect.
(2) Method of measurement.
(3) Correct location of temperature control point.
(4) Effect of variation in wall construction and wall temperatures. This work must lead to a standard procedure or code for the testing of heating effect as defined.

e. That in view of the foregoing, a Committee be appointed to revise and bring up to date in the form of a code, the present exposed radiator test method, having in mind the desirability of correlating this code with that now under consideration on concealed radiation. The Committee further recommends that these test codes be published in separate pamphlets available to all members for insertion in their book of A. S. H. V. E. Codes.

f. That definite tests be authorized to prove whether

or not the so-called hot room method of testing gives the correct results, bearing in mind that if possible, this method should be approved due to its simplicity and low cost. Such a method would be within reach of all manufacturers, while the high cost of the cold room would tend to retard its universal acceptance. Such work is, at present, under way in the University of Illinois, and should be continued on both concealed and exposed heating units.

The Society's research work on heating units is now being carried on under the direction of Professors Willard and Kratz at the University of Illinois in cooperation with the Society's Research Laboratory. Quoting from Professor Willard's report, the following is an outline of the work now under way and planned for the future:

GENERAL STATEMENT—This report presents in a very condensed and summarized form certain data pertinent to the objectives, accomplishments, conclusions, future program, and expenditures for the investigation of radiator performance from the funds provided by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University of Illinois since April 1, 1931. It may be well to point out that the work in progress under the present agreement is, in part at least, a continuation of an investigation in this same field originally sponsored and supported by the *Institute of Boiler and Radiator Manufacturers.*

OBJECTIVES—The investigation as now administered, beginning April 1, 1931, has proceeded with two principal objectives, which are being pursued in two separate and distinct testing plants located in the Mechanical Engineering Laboratory at the University of Illinois.

(1) A continuation of the study, in our room heating testing plant, of the performance characteristics of the various types of cast iron radiators and various types of non-ferrous radiators having heating elements made of copper, brass, and aluminum assembled in enclosures acting as convectors. This plant was erected and completely equipped under the previous cooperative agreement with the *Institute of Boiler and Radiator Manufacturers.*

NOTE: These tests are all being conducted under heating service conditions in actual rooms with outside air temperatures approximately zero degrees and with inside air temperatures taken at the 30-in. level of approximately 68 F. Comparisons between the different radiators tested are then made on the basis of steam condensed per hour and the air temperature gradient between ceiling and floor levels. It should be noted especially that while the air temperature at the 5-ft or breathing line level is alwa s recorded, it is the 30-in. level which is kept the same, at 68 F, and used as the common basis for comparison.

(2) An entirely new study of exposed and concealed ferrous and non-ferrous radiator performance in a special *warm-wall* testing booth. This study has involved the erection and operation of a test booth in accordance with the specifications contained in the "Proposed Code for Testing and Rating Concealed Gravity Type Radiation," Edition of January, 1931, which was adopted in principle at the Annual Meeting of the Society, January 1931.[1]

NOTE: In this test booth the same radiators which have been tested in the room heating testing plant (see item 1) will be again tested in accordance with the "Proposed Code" for the purpose of obtaining correlation factors between the two plants with various types of radiators.

ACCOMPLISHMENTS—Published results of the entire investigation to date appear in three *Bulletins* Nos. 169, 192 and 223, and one Reprint, No. 1, of the Engineering Experiment Station of the University of Illinois, and in three professional papers before the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Only the Reprint and the last paper have been published since April 1, 1931. A verbal report illustrated with lantern slides on the tests in the *warm-wall* booth was made at the Semi-Annual Meeting of the Society, June, 1931. The papers are:

(1) Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, presented at the Annual Meeting of the A. S. H. V. E., January, 1929.

(2) Wall Surface Temperatures, presented at the Semi-Annual Meeting of the A. S. H. V. E., June, 1930.

(3) Steam Condensation an Inverse Index of Heating Effect (see also Reprint No. 1, University of Illinois Engineering Experiment Station), presented at Semi-Annual Meeting of the A. S. H. V. E., June, 1931.

Room Heating Testing Plant

Altogether, about eight types of exposed and concealed cast-iron radiators, and about fifteen types of enclosures for these same radiators have been investigated and their performance characteristics determined in the room heating testing plant under actual winter service conditions. In addition to these tests, about four types of ferrous and non-ferrous concealed convector heaters or radiators have been investigated in a similar manner, and similar performance characteristics determined. These latter tests and some of the former have been made since April 1, 1931, and will be reported on at the Annual Meeting of the Society in January, 1932.

Warm Wall Booth

Two types of convector heaters with non-ferrous heating elements have been tested over a wide range of air temperatures in the *warm wall* booth. In each case, the relation between steam condensation and temperature of the air entering the heater has been established over a wide range of air temperatures. The testing work for establishing such relations cannot progress rapidly because the surrounding air temperatures are determined by conditions in the large laboratory in which the booth is located. After the validity of the results obtained in the *warm wall* booth and the necessary correction factors for deviations from standard entering air temperatures have been completely established, work can progress more rapidly since but one entering air temperature will then be required for each heater tested. Results of these tests comparing the performance of convector heaters with direct radiators in the *warm wall* booth as well as in the room heating testing plant will be reported on at the Annual Meeting of the Society in January, 1932.

CONCLUSIONS—From the results of tests in the room-heating testing plant several important conclusions relative to the performance of various types of radiators in heating rooms have been definitely established.

1. Steam condensation is not a satisfactory index of radiator performance in heating rooms. The *best* radiators condense the *least* steam, not the *most*, for a given *room temperature* and outside temperature.

2. *Room temperature* requires far more specific definition than exists at present, usually understood to mean the average air temperature 5 ft above the floor. Temperatures read at such a level mean very little when comparing widely different types of radiators, and their effect on human comfort in any room.

3. A uniform agreement to read and guarantee *room temperatures* at a level of 30 in. above the floor would establish a satisfactory basis for the comparison of all types of radiators and the *heating effect* which they produce in an ordinary room.

[1] The latest edition of this code appears in the August 1931 issue of *Heating, Piping and Air Conditioning*, p. 695.

4. *Heating effect* is a more or less indefinite and misunderstood term at present since we have no way of measuring it in any usable or truly comparative units. A new basis of evaluation which will reflect the combined effect of radiation, convection, air movement, and humidity on human comfort is needed. The nearest approach to this at present is the eupatheoscope already developed and now in use in England.

5. Radiators of the direct type when *properly* concealed in a given room with a correct enclosure will condense less steam for the same air temperature at the 30 in. level, and maintain a smaller air temperature differential between ceiling and floor than *the same identical radiators* standing exposed.

6. Improperly concealed radiators with incorrect enclosures will not only condense less steam but also will fail to heat the room satisfactorily when using the same identical radiators for both exposed and concealed conditions.

7. The inner surface of the outside walls and glass is always at a lower temperature than the air in a heated room at *any given level,* and the air to wall differential increases very rapidly as the outside air temperature drops. This differential may amount to 12-17 F with zero outside.

8. Human comfort in any room with much outside wall and glass surface is vitally affected by the condition described in item 7. Hence the inside air temperature at the 30-in. or 60-in. level which would produce human comfort in mild winter weather *cannot possibly produce equal human comfort in severe winter weather.* As a result of this fact and the condition named in item 9 following, the inside air temperature at any level must be raised in severe winter weather to maintain the same human comfort as in mild winter weather.

FIG. 6—THIRTY-SEVEN STORY GRANT BUILDING, PITTSBURGH, PENNSYLVANIA, WHERE A SURVEY IS TO BE MADE BY THE RESEARCH LABORATORY ON THE VENTILATING SYSTEM IN THE FIVE-LEVEL BASEMENT GARAGE

9. Since the air temperature differential between ceiling and floor in any room with much outside wall and glass surface increases rapidly as the outside air temperature drops, human comfort based on a high level temperature reading such as 5 ft above the floor will be seriously affected in severe winter weather. This differential may amount to 15 to 20 F with zero outside. The maintenance of a constant comfortable air temperature somewhat nearer the floor, as at the 30-in. level is, therefore, most desirable in heating rooms by either exposed or concealed radiators.

10. The most important conclusion from all of this research work is that tests of all types of exposed and concealed radiators for heating rooms must be made under actual service conditions if any correct conception of the complete performance characteristics and the effect on human comfort of such heating units is to be secured.

Warm Wall Booth

From the results of a very limited number of tests on concealed convector heaters in the *warm wall* booth, the following observations or tentative conclusions have been drawn:

1. Based on the data so far obtained, the evidence supports the conclusion that the steam condensations obtained in the *warm wall* booth are valid, in that they are in agreement with those obtained in the room heating testing plant when run with the same temperature for the air entering the heater, or when the proper correction factor is applied to reduce the results to a common entering air temperature.

2. The results so far also indicate that over a fairly wide range of entering air temperatures the heat output of the convector heaters varies as the 1.3 power of the temperature difference between the steam and the entering air with sufficient accuracy to justify the use of this relation for the correction to standard entering air temperature if the deviations from the standard temperature are kept within reasonable limits.

FUTURE PROGRAM—It is quite apparent that one of the most important fields for future study in connection with radiator room heating has been indicated already in Item 4 of the Conclusions.

Some progress has been made in designing and assembling auxiliary apparatus for a eupatheoscope, or instrument for measuring the comfort conditions in a room as evaluated by the heat loss from a copper cylinder with the surface temperature maintained at 75 F. Some experimental work has been done to determine the most feasible arrangement of thermocouples and electrical heaters to be used in connection with this cylinder, as well as the adaptability of the recorders available. This eupatheoscope will prove a valuable supplement to the temperature gradients in the rooms in studying the heating effects of various types of radiators.

It is also probable that a rearrangement of the rooms in the present testing plant should receive very serious consideration. Undoubtedly, confirmatory tests using radiator and enclosure equipment already tested should be made in (1) large rooms of the same general proportions and characteristics, and (2) rooms with a single outside wall exposure instead of two outside walls. For these latter tests, the larger room is quite essential in order to secure adequate heat loss with only one wall.

In the case of the *warm wall* booth, the future testing program should include further tests on exposed and concealed cast-iron steam radiators, more tests on convector

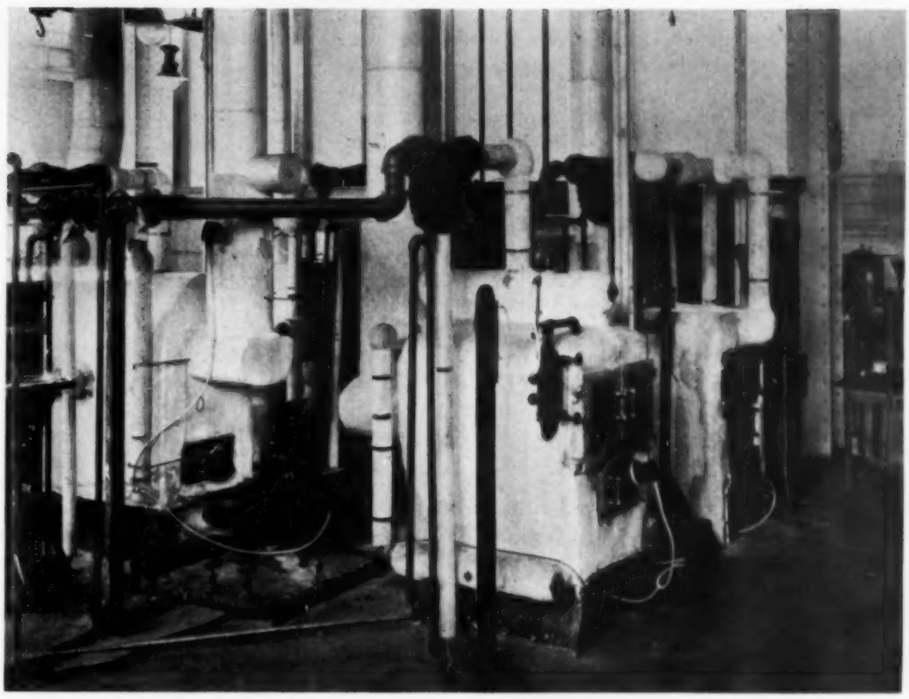


FIG. 7 — GENERAL VIEW OF SET-UP OF OIL BURNER TEST APPARATUS AT YALE USED IN OBTAINING DATA FOR A STUDY OF PERFORMANCE CHARACTERISTICS OF OIL BURNERS AND LOW PRESSURE HEATING BOILERS

heaters with steam, and finally, a limited series of tests on all types of radiators with hot water.

6. GARAGE VENTILATION

Ventilation of Garages and Bus Terminals

Technical Advisory Committee
E. K. Campbell, *Chairman*
D. S. Boyden
H. P. Gant
W. T. Jones
W. C. Randall

The Technical Advisory Committee has given a great deal of consideration during the past year to the continuation of the study of garage ventilation. Complete plans were recently perfected for a study to be made by the Laboratory in the five-level basement garage of the 37-story Grant Building in Pittsburgh. This work will be undertaken at once.

7. HUMAN COMFORT

Air Conditions and Their Relation to Living Comfort

Technical Advisory Committee
C. P. Yaglou, *Chairman*
O. W. Armspach
W. L. Fleisher
Dr. E. V. Hill
Dr. R. R. Sayers

Investigations under this Technical Advisory Committee have been carried on by the Society's Research Laboratory in Pittsburgh and by Professor Yaglou of the Harvard School of Public Health during the past several years.

a. During the past year the Laboratory completed the study of heat loss from the bodies of men working and presented a paper to the Society on this subject.

b. The Laboratory has since completed the study of heat and moisture loss from the bodies of children of school age and the verification of application of effective temperature scale to school house ventilation. A paper on this subject was presented at the annual meeting of the *American Public Health Association* in Montreal in September, 1931.

c. The Laboratory has under way in Pittsburgh the study of the effect of cold wall surfaces on the desirable effective temperature comfort.

As soon as work now under way in the psychrometric chambers in the Laboratory in Pittsburgh is completed, the following two projects will be understaken: (*a*) A study of the physiological and bacteriological effects of humidity and changes of humidity on the human organisms. (*b*) A study of physiological and bacteriological effects of cold and the effects of drafts, low temperature, etc. on the human organism.

Under the general heading of the study of vital characteristics of the atmosphere, Professor Yaglou of the Harvard School of Public Health in cooperation with the Research Laboratory has been working on the following three investigations:

a. A study of the differences in ionic content in outdoor air and expired air, and the effects of room occupancy upon ionic content.

b. De-ionizing effect of air-conditioning methods and apparatus.

c. A thorough study of diurnal and seasonal variations in atmospheric ionization, both out of doors and indoors, in Boston, and in at least three other cities having different climates. Observations to be correlated with corresponding variation in the incidence of seasonal diseases.

Items *a*, *b* and part of *c* were approved by the Research Committee early this year and are now being studied. A progress report on the changes in ionic content of air in occupied rooms and on the de-ionizing effects of air-conditioning methods and apparatus has already appeared in the October issue of *Heating, Piping and Air Conditioning* and will be presented to the Annual Meeting of the Society, January 1932.

The Committee on Research recently approved the study of subject *c* to cover at least three other localities, namely Pittsburgh, Denver, and one or two other cities on the Pacific Coast.

The Technical Advisory Committee has recommended the following five subjects for future investigation:

a. A study of the growth of organisms and insects in air-conditioned rooms and in air washers, Baudelot coolers, etc., with the object of finding a water soluble, non-toxic, element for checking or preventing growth of lower organisms.

b. The effect of space charge (ionic unbalance) on the colloidal state of certain substances (milk, yeast, glue, and other common gelatinous substances).

c. A thorough study of air requirements for ventilation

both in winter and summer under customary seasonal range of room temperature and humidity. Experiments to be carried on in a conditioned chamber using an ideal laboratory system of distribution and a temperature rise of 10 to 30 F in the summer experiments.

d. A study of air distribution with the object of improving present systems, or developing new and more efficient systems.

e. Organization of a new committee on Refrigeration in Relation to Air Conditioning for the purpose of disseminating knowledge and sponsoring research.

8. OIL HEATING

Oil and Gas Burning Devices

Technical Advisory Committee

L. E. Seeley, *Chairman*
R. M. Conner
P. E. Fansler
R. S. Franklin
R. V. Frost
R. C. Morgan
H. F. Tapp

A study of methods of testing oil burner boiler combinations is being carried on under the direction of Professor Seeley of Yale University in cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Oil Burner Association.* The investigation has for its purpose the development of a standard code for testing oil burners—boiler or furnace combinations. As a result of the work so far completed, there has been prepared and submitted to the Society a Code for Testing Low-Pressure Steam Heating Boilers Burning Oil Fuel[3]. Also, a complete report of a study of sixteen combinations of different boilers and burners showing (*a*) effect of various fuel burning rates upon quality of combustion (*b*) effect of draft fluctuation upon combustion (*c*) the effect of quality of combustion upon the efficiency and upon the flue gas temperature, (*d*) characteristic efficiency curves, (*e*) influence of heating surface upon efficiency, (*f*) influence of burner type on efficiency, (*g*) sound measurements, (*h*) power requirements of oil burners, (*i*) actual heat liberation rates per cubic foot of furnace volume, (*j*) characteristic efficiency curves due to intermittent burner operation, etc.

A paper will be presented at the Annual Meeting of the Society, January 1932 giving the results of a more intensive study of intermittent burner operation. This matter is a practical one since it represents what actually occurs in practice and will look towards constructive steps in the proper application of boiler and boiler load for best results on intermittent operation as well as considering means of control (*i. e.,* pressure or thermostatic).

A study is being made of possible methods of testing hot water and vapor installations. It is felt that the type of heating system also has some influence upon overall economy. In the event of satisfactory test methods being worked out, there will then be a basis for the development of two test codes.

9. SOUND

Sound in Relation to Heating and Ventilation

Technical Advisory Committee

F. B. Rowley, *Chairman*
Carl Ashley
Warren Ewald
R. F. Norris
G. T. Stanton

This is a new project given consideration by the Committee on Research and the Technical Advisory Committee during the past year. No work has been undertaken in the Laboratory. The Technical Advisory Committee is studying the problem under the following three headings:

a. The selection or development of a standard method and standard equipment for measuring sound. This must be accurate, reliable and should be reasonably cheap and convenient for field operation.

b. A study of the causes of sound in ventilating equipment and a better understanding of methods of eliminating sound at its source.

c. A study of the methods of reducing the effect of unavoidable sounds and noises to unobjectionable levels.

[3] See *Heating, Piping and Air Conditioning,* April 1931.

10. AIRATION

Infiltration in Buildings

Technical Advisory Committee

G. L. Larson, *Chairman*
J. E. Emswiler
F. E. Giesecke
W. C. Randall
W. A. Rowe
J. G. Shodron
Ernest Szekely
M. S. Wunderlich

With the completion of the studies of infiltration through built-up wall sections of different types of construction, the future study of infiltration has been directed towards other phases of the general question of airation of buildings. Five projects have been approved by the Technical Advisory Committee. Work on some of these projects was carried on during 1931 under the direction of Professor Larson at the University of Wisconsin in cooperation with AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Laboratory. The committee has recommended that other work proposed by it be carried on at the laboratory in Pittsburgh and elsewhere. The following are the five projects as prepared by the Technical Advisory Committee:

a. STUDY OF DOUBLE HUNG WOOD WINDOWS.

Following the suggestion made in the last paragraph of the paper on Air Infiltration Through Double-Hung Wood Windows as reported at the Semi-Annual Meeting, June 1931 that the windows of that program be exposed to wear and weather, a shelter house has been built for that purpose. The fourteen windows of the test program are in the weather-stripped condition, nine with an average crack and clearance and five with a fit representing poor conditions.

The windows have been carefully painted on the outside exposed surface and stained and varnished on the inside surface in the same manner as would be done in house construction. Each window sash was opened and closed thirty-five times so as to free the sash perimeter surfaces of surplus paint. The windows were then retested for air leakage before installation in the shelter house where they are exposed to ordinary weather on the outside and the inside will be at 70 F. The framing of the house is made in such a way that the windows can be installed or removed for further testing with the minimum of effort. The temperature will be maintained at 70 F by thermostatic control.

It is proposed to open and close each sash a definite number of times each week to provide an element of wear present in actual building constructions. At intervals the pulls required to operate the sash will be taken and recorded. At any time it is deemed advisable the window may be removed and tested for air leakage in the laboratory. It is proposed to do this at the end of each year of exposure.

The pull to raise and lower the sash has considerable importance. Weatherstripping is often resorted to to make a sash that works freely and yet does not rattle. The original plain windows of this program worked very freely. The installation of the weatherstrips increased the sash pulls but the windows were still very easily operated. It would seem that since a weatherstrip increases the sliding contact surface area that it must necessarily increase the sash pulls over those for the freely operating plain window. The exposure tests will determine whether this increase in required sash pulls is permanent or whether it will decrease with the length of time of exposure.

Another question of interest is to determine whether the three different types of weatherstrips represented maintain their efficiency in preventing air leakage equally well. There is a feeling among some weatherstripping mechanics that certain types of strips can be fitted to show up well on strictly laboratory tests, but that upon exposure to wear and weather, they will not maintain a high efficiency.

b. EFFECT OF ELIMINATING VENT OUTLETS IN SCHOOL ROOM VENTILATION, OR EXFILTRATION IN VENTILATED AREAS.

Tests have been made on the ventilation of a typical class room in one of the Madison, Wis., public schools. The study was undertaken to determine the effect of outlet ducts in providing ventilation that is proper in amount and distribution.

This school building is equipped with unit ventilators equipped with d-c motors so that the output in cubic feet of air may be readily changed. Anemometers were used to determine the air quantities at the unit ventilator and at the outlet grille. The anemometers were carefully calibrated by moving at various speeds over the circumference of a 26-ft diameter circle with an electrical means of starting and stopping the anemometer while in motion.

A duplicate unit ventilator was set up in the laboratory to which air was supplied by a motor-blower through a duct work. The purpose of this set-up was to calibrate the anemometer for use on the grille of the unit ventilator. Similarly a duplicate outlet duct and grille was set up in the laboratory and the anemometer calibrated for use against this grille. A Pitot tube was used for air measurement in these calibrations.

The tests in this school room showed that about 55 per cent of the air entering the room through the unit ventilator was discharged through the outlet duct and grille under normal operating conditions. When every visible crack in the room was sealed, the capacity of the unit ventilator was reduced to 72 per cent of its normal capacity. In this test the outlet duct, the door and transom, and the sash perimeters of all windows were sealed. In addition all large plaster cracks and joints in the mullions were sealed. When the seal was removed from the door and transom leading to the corridor, the air quantity increased to 85 per cent of normal capacity. The removing, in addition to this, of the seal on the sash perimeters of the windows increased the air delivery of the unit ventilator to 92 per cent of the normal capacity as found with the outlet duct open. Tests are to be made under winter heating conditions to determine the influence of outlet ducts on the air distribution within the room.

Similar tests are being made on a 23-ft by 32-ft class room in the new Mechanical Engineering Building of the University of Wisconsin. The classrooms in this building are ventilated by a central fan system. In the tests made so far the regular inlet to the room from the central fan system has been kept closed. A motor-blower has been set up to draw in air from outside through a duct equipped with a Pitot tube for air measurement. The use of the motor-blower set-up makes it possible to measure the air coming into the room accurately and further eliminates the influence of the other ventilated rooms on the ventilation of the room under study. The central fan system was kept in operation during all tests so as to maintain normal conditions in the corridor into which the class room air is discharged. The air delivery has been measured under various conditions, such as with the outlet louvers in the door, the perimeter of the windows and door, sealed and not sealed.

Tests when approximately 1200 cfm are being supplied to this class room by the motor-blower check the results found for the city school class room. When the outlet louvers were sealed, the air volume was reduced to 92 per cent of the normal amount. Sealing the door perimeter in addition reduced the air volume to 85 per cent of normal. Sealing the window perimeter so that all visible openings were sealed, reduced the air volume to 74 per cent of normal. The determination of the amount of air passing through the louvers in the door which constitute the outlet opening for the room awaits the determination of the proper calibration coefficient for the anemometer when used under these conditions. This is to be done in a laboratory set-up where the air may be measured by a Pitot tube. This, of course, cannot be done in the class room since it seems impossible to find and stop all exits for air.

The drop in pressure in inches of water from the room to the corridor were as follows: for normal operation, 0.018 in., for the louvers sealed, 0.320 in., for the louvers and door perimeter sealed, 0.578 in., and for the louvers, door and window perimeter sealed, 1.020 in. This shows considerable building up of pressure and accounts for the maintenance of a high air delivery as the openings available for exfiltration are reduced.

FIG. 8—GROUP OF REPRESENTATIVE SCHOOL CHILDREN IN PSYCHROMETRIC CHAMBER AT THE A. S. H. V. E. RESEARCH LABORATORY. CERTAIN PHYSIOLOGICAL REACTIONS ARE BEING DETERMINED UNDER VARIOUS ATMOSPHERIC CONDITIONS WHICH THEY MIGHT ENCOUNTER IN THEIR HOMES, SCHOOLS, OR AT THE THEATER. A PAPER RESULTING FROM THIS STUDY WAS PRESENTED BEFORE THE AMERICAN PUBLIC HEALTH ASSOCIATION DURING ITS CONVENTION IN MONTREAL, CANADA. IN THE NEAR FUTURE, THE RESULTS OF THE STUDY WILL BE PRESENTED TO THE A. S. H. V. E.

The results obtained no doubt depend on the characteristics of the system used to supply the air. In any particular case, that is, with the use of a central fan, a unit motor-blower or a unit ventilator the influence of the various outlets for air depends on the characteristics of the air supply unit as to the change in air delivery with a change in delivery pressure. Results obtained with a motor-blower set-up probably apply only to such a set-up, and more particularly to a motor-blower set-up having the same characteristics of air delivery and discharge pressure as the one used on the tests. This factor likely is more serious with unit ventilators than with fan systems either unit or central in type that are designed to work against a considerable resistance.

A unit ventilator is to be set up in this same class room and the delivery of air under various conditions of sealing determined. Quite likely the sealing of various air outlets will have a greater effect than in the case of the motor-blower set-up. This room and the building in which it is located are of the best and of new construction. The only evident places for air to get out of the room are around the overhead supply main at the wall partitions into adjoining class rooms, around two ¾-in. return risers leading to the main laboratory space, and around the trim of the door. The openings around the steam main are only effective when the pressure in the class room under test is higher than in adjoining class rooms, which likely happens only under test conditions with some of the normal openings sealed. The pipe covering on the main fits against the plaster walls very tightly so that no opening is visible at these locations.

Tests will also be made with the central fan system supplying air to the class room under test. It will be necessary to determine the proper coefficient for an anemometer for the inlet grille in this case. The tests of a single room are presumed to present typical results for all of the rooms connected to a central system. When tests are made with the outlet louvers sealed or the door or window perimeters sealed, it would seem necessary to apply the same sealing to all of the other rooms supplied by the central fan system. If this is not done, the building up of resistance in the room under test would cause a shift of air delivery to the other rooms. This complicates greatly the testing when the central fan system is used to supply air to the test room. Since the motor-blower used on the first series of tests is a pressure fan very similar in type to the central fan, likely this first series gives a close determination of the behavior to be expected with a central fan system.

The motor-blower set-up is not arranged to deliver the air into the room in a uniform way so that air distribution tests will not be made. Such tests will however be made with the unit ventilator set-up and the central fan system supplying air to the room.

c. AIR DRIFT AROUND BUILDINGS.

This project has been discussed in previous reports. The phase concerned is the effect of outside wind conditions on so called *window ventilation,* particularly relating to schools.

Readings are being taken to determine the variation in the amount of air entering through an open window at various wind velocities. Two class rooms of about the same size have been selected on the second floor, one on the north side and one on the south side. The building is four stories high but has a fully exposed and heated basement with outside openings. It is considered that the rooms are near the center of the height of the building, so that temperature effect is as small or smaller than it would be at any other place in the building. The inlet ducts are kept closed. The outlet ducts so far have been kept open.

One window in each room is opened a definite height such as would be used for open window ventilation. An anemometer is set in each opening and readings observed at five-minute intervals. At the end of one-half hour, anemometers are exchanged and readings continued another one-half hour so as to eliminate any differences that may be due to anemometers. Corrections are applied to the anemometers, but of course if the velocity is very low the reading is not accurate.

The wind velocity and direction of the wind is obtained from the weather station located on the roof of the next building. The wind velocity is obtained in direction for each minute of every five minutes and in velocity over the five-minute periods. The assumption is made that only the wind velocity normal to the wall surface affects the inflow of air, and the velocity in directions other than normal are reduced to the normal component. Since there might be a time lag between the wind velocities at the weather station and at the windows concerned, the one-half hour averages are used for plotting.

Only in one case was it found that air was flowing out through the window. This happened when the outside temperature was 81 F. In all cases, there was an inflow of air on the leeward side due to the temperature effect.

If the rough assumption is made that the difference in the two inflow velocities is the effect of the wind, the points very roughly determine a straight line passing through zero. This shows that every mile of wind velocity causes an increase in inflow velocity on the windward side of 40 fpm. Again, if the rough assumption is made that the inflow on the leeward side is all due to temperature effect, the points very roughly determine a line passing through zero that gives an increase in inflow on either side of 100 fpm for each 10 deg temperature difference.

The variation in inflow and outflow openings in the building as a whole affect the neutral zone. These openings change with the outside temperature. Likely there were many openings when the outside temperature was 81 F. At the same time, probably the temperature inside was higher than 70 F. The wind is extremely variable in direction and velocity and the weather station figures and directions likely do not give the correct readings at all times. For instance, the wind if it were steady between north and northeast for one minute would be recorded as one or the other, whereas, actually the velocity was between the two. The anemometers, although kept well cleaned and oiled and reserved for this use, are not accurate at low velocities. The readings during the winter will be mostly with a north wind and a large temperature difference. It is highly improbable that an outflow will be found on the leeward side during this season.

As a variation of the foregoing procedure, the inflow of air with the outgoing ducts blocked is to be tried. Undoubtedly, the inflow through the leeward window is due largely to the large area permitting air to leave the room in the outgoing ducts. This, of course, needs a temperature difference to act, unless there is an aspirating effect from the air leaving other parts of the building through the outlet ducts. A trace of air currents with open window ventilation would be of interest to show velocities and distribution through the rooms. Likely, there is a shorting along the floor to the outlet duct.

The present plan is to complete the series with the outlet duct blocked and incorporate the results in a paper on Effect of Eliminating Vent Outlets in School Ventilation.

d. AIR STRATIFICATION IN CONFINED SPACES AND THE EFFECT OF CROWDS AND AIR MOVEMENT ON THE SAME.

This study was begun last winter in the new University of Wisconsin Field House. The height of this indoor stadium is approximately 100 ft from floor to peak of roof. The building is designed to eventually seat about 12,000 people and at present accommodates about 8000. There is a difference in altitude of about 45 ft between the lowest and highest rows of seats.

Test readings were made at all the basketball games during the winter of 1930-31. Temperature readings were taken at 10-ft intervals from floor to roof and also at various rows of seats. Humidity readings were taken with a recording instrument. Unfortunately last winter was an unusually mild one in Wisconsin, and the coldest temperature at any basketball game was about 20 F above zero. It is planned to continue this work

FIG. 9 — STAFF AT A. S. H. V. E. RESEARCH LABORATORY, PITTSBURGH, PENNSYLVANIA. FROM LEFT TO RIGHT: PAUL MCDERMOTT, CARL GUTBERLET, A. E. KURTZ, F. C. HOUGHTEN, J. L. BLACKSHAW

during the 1931-32 basketball season in the hope that it will be possible to study the conditions that will arise when the outside temperature is around zero and below.

e. INFILTRATION AND EXFILTRATION IN TALL BUILDINGS.

This is a project the committee has had under consideration for some time, and some data are available upon the subject. It is proposed that this study be undertaken by the Laboratory in Pittsburgh on some tall building.

11. PUMPS AND TRAPS

Devices for Handling Condensation and Air

Technical Advisory Committee

E. K. Lanning, *Chairman*
W. E. Barnes
C. A. Dunham
I. C. Jennings
F. J. Linsenmeyer
J. C. Matchett
A. W. Moulder
E. J. Ritchie
W. K. Simpson
H. G. Thomas
R. H. Thomas

This is a new project considered by the Committee on Research during the past year. The Technical Advisory Committee is studying the problem with a view of outlining its needs.

12. REFERENCE DIRECTORY

Correlating Thermal Research

Technical Advisory Committee

R. M. Conner, *Chairman*
D. S. Boyden
P. D. Close
J. C. Fitts
W. T. Jones
H. T. Richardson
Perry West

At the last annual meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held in Pittsburgh January 26 to 29, 1931, the Research Committee received a proposal from the Directors of the *Association for Correlating Thermal Research* that the card catalogs of available literature developed by the Association together with the physical property of the Association and the funds and pledges in its treasury be taken over and administered as a function of the Research Laboratory.

The Committee on Research agreed to take over the property and activities of the Association at the close of the present year, with the understanding that the Association continue to function under its Board of Directors and its own finances during the year. The Technical Advisory Committee on Correlating Thermal Research of the Committee on Research was appointed to look after the Research Laboratory's interest in the undertaking during the current year, and to receive the property and administer the work after the transfer to the Laboratory had been made. This transaction will be consummated before the annual meeting of the Society.

The annual Directory was published during the year. In getting out the new Directory, much wider cooperation was enjoyed from other associations, institutions and universities than ever before, with a result that the current Directory is more comprehensive and valuable and the influence of the Association in correlating research in this branch of the engineering industry has been extended. In developing cooperation with allied organizations, a deeper appreciation on the part of other organizations was noticed.

The Librarian of the Association continued to receive and answer inquiries for information pertaining to published literature and research activities. This branch of the work of the Association, which is the real object for which the work was originally undertaken, is being more fully realized. Copies of the Directory may be had by addressing the A. S. H. V. E. Laboratory, 4800 Forbes St., Pittsburgh, Pa.

National Research Council Organization and Members

A pamphlet has recently been issued by the National Research Council giving its organization and members for 1931 and 1932. The National Research Council is a cooperative organization of the scientific men of America. Its members include, however, not only scientific and technical men, but also business men interested in engineering and industry. It was established in 1916 by the National Academy of Science and enjoys the cooperation of the major scientific and technical societies of the country, its membership being largely composed of 77 of these societies, representatives of certain other research organizations, representatives of government scientific bureaus, and a limited number of members at large. The Society is represented on several of the committees and subcommittees.

Among the various methods of contributing practical assistance to American science in harmony with the general point of view which the National Research Council has so far adopted, perhaps the most important is the establishment of special committees of carefully chosen experts for specific scientific subjects or problems urgently needing consideration, who shall plan modes of attack and undertake to find men and means.

PROGRAM

38th Annual Meeting

American Society of Heating and Ventilating Engineers

Hotel Statler **January 25-29, 1932** **Cleveland, Ohio**

Monday—January 25th

8:00 A. M.—Finance Committee
9:00 A. M.—Committee on Ventilation Standards
10:00 A. M.—Meeting of the Council (*Parlor C*)
12:00 Noon—Registration (*Mezzanine*)
12:30 P. M.—Luncheon for Officers and Authors (*Parlor G*)
1:30 P. M.—Meeting Committee on Research (*Parlor O*)
2:00 P. M.—Opening of Second Heating and Ventilating Exposition (*Cleveland Auditorium*)
7:00 P. M.—Joint Dinner-Meeting of the Councils of the A. S. H. V. E. and *A. S. R. E.* and their wives (Hotel Cleveland)
8:00 P. M.—Committee on Radiation (*Parlor R*)

Tuesday—January 26th

9:30 A. M.—First Session (Ball Room Hotel Statler)
Address of Welcome
Response by Pres. W. H. Carrier
Reports of Officers
Report of Council
Reports of Council Committees
Membership, Roswell Farnham, *Chairman*
Publication, W. A. Rowe, *Chairman*
Report of Committee on Research, C. V. Haynes, *Chairman*
Report of Director of Research Laboratory, F. C. Houghten
Reports of Special Committees
Guide Publication Committee, D. S. Boyden, *Chairman*
Committee on Increase of Membership, C. W. Farrar, *Chairman*
Reports of Continuing and Cooperating Committees
Report of Tellers of Election, F. A. Kitchen, *Chairman*
Technical Papers:
Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and W. A. Eckley
Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet
Transmission of Radiant Energy Through Glass, by R. A. Miller and L. V. Black

2:00 P. M.—Second Session (Ball Room, Hotel Statler)
Technical Papers:
Importance of Radiation in Heat Transfer Through Air Spaces, by E. R. Queer
Heat Emission from Iron and Copper Pipe, by F. C. Houghten and Carl Gutberlet
Hot Water Friction Heads in One-Inch Cast Iron Tees, by F. E. Giesecke and W. H. Badgett
Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford

7:00 P. M.—Inspection of Second Heating and Ventilating Exposition
9:00 P. M.—Informal Reception including entertainment and Monte Carlo (Ball Room Hotel Statler)

Wednesday—January 27th

10:00 A. M.—Third Session (Little Theatre, Cleveland Auditorium)
Joint Meeting with *A. S. R. E.*—Program arranged by A. S. H. V. E.—Pres. W. H. Carrier presiding
Technical Papers:
Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate
Acoustical Problems in Heating and Ventilating of Buildings, by V. O. Knudsen
Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, Paul McDermott and E. M. Pugh
Field Studies of Office Building Cooling, by S. S. Sanford, E. P. Wells and J. H. Walker

1:30 P. M.—Trip to Akron (Goodyear-Zeppelin Hangar) with *A. S. R. E.*
(Buses leave from Hotel Statler)
7:00 P. M.—Dinner for Past-Presidents and Wives (*Tavern Room*, Hotel Statler)
8:00 P. M.—Chapter Relations Committee
Ladies' Theatre Party at Keith's Palace
8:00 P. M.—Inspection of Second Heating and Ventilating Exposition

Thursday—January 28th

10:00 A. M.—Fourth Session (Little Theatre, Cleveland Auditorium)
Joint Session with *A. S. R. E.*—Program arranged by *A. S. R. E.*—Alvin H. Baer presiding
Technical Papers:
Application of Electric Refrigeration to the Heating and Cooling of Houses, by F. H Faust, E. W. Roessler and A. R. Stevenson, Jr.
Air Conditioning Applied to Cold Storage Including a New Psychrometric Chart, by C. A. Bulkeley
Biological Aspects of Thermal Engineering, by Samuel R. Prescott

12:30 P. M.—Luncheon Meeting
Nominating Committee
2:00 P. M.—Fifth Session (Ball Room, Hotel Statler)
Report of the Committee on Testing and Rating Unit Ventilators, John Howatt, Chairman
Report of Committee on Ventilation Standards, W. H. Driscoll, Chairman
2:30 P. M.—Ladies' Bridge (Chamber of Commerce)
7:00 P. M.—Annual Banquet and Dance (Ball Room, Hotel Statler)

Friday—January 29th

10:00 A. M.—Sixth Session (Ball Room, Hotel Statler)
Technical Papers:
A Study of Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers
Room Warming by Radiation, by A. H. Barker
Comparison of Performance of Conductor Heaters, by A. P. Kratz
A Study of the Combustible Nature of Solid Fuels, by R. V. Frost
Installation of Officers
Resolutions
Adjournment

12:30 P. M.—Luncheon and Meeting of Council (*Tavern Room*, Hotel Statler)
2:30 P. M.—Inspection Trips to Cleveland Industrial Plants

Local Chapter Reports

Cleveland

December 2, 1931. The December meeting of the Cleveland Chapter was called to order in the rooms of the Engineering Society by Pres. W. E. Stark.

Announcements received from A. V. Hutchinson, secretary of the Society, regarding the membership, employment and other Society activities were read, and these caused considerable discussion.

President Stark spoke briefly on limited membership and presented to the Chapter the recommendations of the Board of Governors for this class of membership. It was moved and seconded that steps be taken to obtain from the parent body approval of these recommendations in order that they may be put into operation. The motion was carried, and then President Stark brought out the point that members had two things to sell—membership in the Society, and failing in that—limited membership.

He also spoke on the unemployment situation, mentioning the joint meeting of representatives of the fourteen sections of National Technical Societies whose aim is to organize a joint committee on technical employment.

T. A. Weager, *General Chairman* for the Annual Meeting, told of some of the arrangements that were being made for entertainment during the meeting.

The speaker of the evening was J. C. Miles, who spoke on the subject of Air Conditioning the American Home. He expressed the opinion that the systems of heating that were being designed at present were out of date and that the future would see an increase in the use of convection heating with conditioning of the air.

An interesting and spirited discussion followed the talk, and a vote of thanks was extended to Mr. Miles.

Western New York

January 11, 1932. Pres. Joseph Davis received reports from the various committees which had functioned during his administration, and thanked all for the hearty support given him during the past year.

An unanimous vote for the election of officers for the coming year was recorded as follows:

President—M. C. Beman
1st Vice-President—D. J. Mahoney
2nd Vice-President—J. J. Yager
Treasurer—W. F. Johnson
Secretary—W. E. Voisinet

An enthusiastic discussion of the Annual Meeting of the Society to be held in Cleveland was started by Roswell Farnham and practically the entire Chapter signified its intention of attending.

H. M. Hammond, Bailey Meter Co., was the speaker of the evening, and talked on The Use of Recording Instruments in the Boiler Room. With the aid of interesting and descriptive slides, Mr. Hammond gave a very instructive and entertaining talk on metering devices covering a wide range of typical boiler plants. Measurement and charting of steam, draft, feedwater, etc., were clearly described and it was shown how the graphic relation as indicated was used to cut down boiler costs. A discussion followed the address, after which a vote of thanks was given to Mr. Hammond.

December 14, 1931. The Western New York Chapter of the Society held their annual Christmas stag party at the University Club, where 40 members and guests enjoyed bowling and cards, the party being in the form of a Monte Carlo.

A short business meeting was presided over by Pres. Joseph Davis, and the following slate of officers were announced for the coming year: *President,* Myron C. Beman; *First Vice-President,* D. J. Mahoney; *Second Vice-President,* J. J. Yager; *Treasurer,* W. F. Johnson; *Secretary,* J. J. Landers.

President Davis was named as representative and Mr. Mahoney as alternate to serve on the Nominating Committee at the Annual Meeting to be held in Cleveland, January 25-29, 1932.

Minnesota

December 21, 1931. The Minnesota Chapter of the Society held its regular meeting at the Washburn High School, with 61 members and guests present, and Pres. H. E. Gerrish presiding.

Following an excellent dinner, a short business meeting was held, at which time President Gerrish introduced some new members.

Albert Buenger, as chairman of the Membership Committee, gave a report on the membership in the Chapter. Prof. F. B. Rowley was then called upon and spoke on Society activities.

C. E. Lewis was appointed as representative on the Nominating Committee, with A. J. Huch as alternate.

The meeting was then turned over to A. L. Sanford, engineer for the Board of Education. Mr. Sanford described briefly the mechanical features of both the Washburn and Ramsay schools, and then introduced the speakers of the evening, Paul Smith, Industrial Welding Co., who gave a short talk on welding, with reference to some of the welding features used in the Ramsay school and E. T. Bros, Wm. Bros Co., who spoke on stokers, dealing with the development of stokers, and also, the cost of operation as compared with other fuels.

H. S. Morton, Stott Stoker Co., also gave a short talk on stokers, dealing with the engineering data required and the method of selecting a stoker for the required load.

Following these talks an inspection was made of the boiler room which carries the heating load for both schools, then a trip through the tunnel from the Washburn High to the Ramsay Junior High, and a preview of the Ramsay Junior High which is to be opened in January.

Illinois

January 11, 1932. The January meeting of the Illinois Chapter, held in the Crystal Room of the Hotel Sherman, was officially designated as "Ladies Night" and 27 women were present in addition to 63 men. As a feature of the evening, special music and entertainment was provided by the Chapter.

Pres. J. J. Aeberly announced the death of James H. Davis on January 5, 1932. Mr. Davis was a Charter and Life Member of the Society and one of its oldest members, having joined with the original group in 1895 and serving on the Council in 1911.

John F. Hale presented the following resolution and it was unanimously voted that this should be spread upon the minutes and a copy sent to the family of Mr. Davis:

Whereas, James H. Davis was a Charter Member of the American Society of Heating and Ventilating Engineers and also of the Illinois Chapter, and

Whereas, he was a valuable and untiring member during the years of his activity and is recalled by a host of friends for his personal charm and good fellowship, and

Whereas, his spirit has passed on to enjoy his unity with Almighty God, be it

Resolved, that we, his friends and associates, hereby express our gratification of having known him, and be it further

Resolved, that this resolution be spread upon our records and a copy be sent to his widow and family, that they may know with what high esteem he was held in our hearts.

An announcement was made by President Aeberly that due to a misunderstanding about the procedure at the December meeting in connection with the amendment to the Constitution and By-Laws, the committee, which prepared the amended Constitution, and a committee of those protesting the December action had met several times and had been appointed as a standing committee to continue to work on the matter. President Aeberly said that no definite action could be taken until the Council had rejected or approved the form of Constitution passed at the December meeting and submitted to them.

In introducing the speaker of the evening, Loran D. Gayton, assistant city engineer, President Aeberly mentioned briefly the scope of the work in which Mr. Gayton is interested, the magnitude of the project on which he is working, and his ability as an engineer and a speaker.

Mr. Gayton then addressed the meeting on the subject of Chicago as a Seaport and showed specifically just what the *deep waterway to the sea* proposed to do and what vital importance it should hold to every citizen of the Middle West.

At the conclusion of the address, the meeting was given to an open discussion in which Mr. Gayton capably answered the questions of his audience.

It was regularly moved and seconded that a written vote of thanks be sent to Mr. Gayton for his address.

December 14, 1931. The usual dinner preceded the meeting of the Illinois Chapter, held at the Hotel Sherman, Chicago, where 53 members and guests were in attendance.

The meeting was called to order by Pres. J. J. Aeberly, who announced the following committee assignments: *Better Relations Commission,* H. G. Thomas, *Chairman,* R. B. Hayward and H. R. Linn; *Membership Committee,* C. W. DeLand, *Chairman,* G. H. Blanding and J. J. Haines; *Legislative Committee,* H. M. Hart, *Chairman,* E. V. Hill and E. P. Heckel; *Representative National Nominating Committee,* A. B. Martin.

President Aeberly then announced that the next question before the Chapter was the adoption of the revised constitution and as he understood it there was some opposition to some of the clauses. He suggested a vote be taken on the revised constitution as submitted and unless two-thirds of the votes were cast in favor of adoption there would be no necessity of submitting the proposed amendments. M. S. Good then explained that there were two changes suggested by A. V. Hutchinson, secretary of the Society, which the committee would like to have made. The changes were the omission of the word *Associate* in the third line of Section II *Article* 3 and the substitution of the words *Publication Committee* for the word *Council* in the fifth line of Section VI of *Article* 6. These changes, it was stated, were necessary to conform to the national Constitution. A vote was then taken and those in favor of adopting the constitution with the changes stated by Mr. Good were asked to raise their hands, and those opposed were asked to do the same. The count showed 20 in favor and 8 against.

J. F. Hale then took the floor and read the following resolution:

Since there is immediate danger that the personnel of the Division of Heating, Ventilation and Industrial Sanitation of the Department of Health of the City of Chicago will be materially reduced, and since such actions by the City Council of the City of Chicago will greatly hamper the heating and ventilating industry, be it

Resolved: That the Legislative Committee of the Illinois Chapter of the American Society of Heating and Ventilating Engineers take prompt action to make known to the responsible persons or parties of the City of Chicago, the protest of this Chapter against such actions, and be it further

Resolved: That the Legislative Committee take such steps as it deems necessary to obtain the full support of other associations and groups within the heating and ventilating industry against this proposed move which, in our opinion, seems unwise and uneconomical at this time.

Mr. Hale explained the emergency and moved that the Legislative Committee be instructed to prepare a resolution along these lines and present it to the proper authorities at the City Hall. R. E. Hattis seconded the motion.

The situation confronting the personnel of the division was then discussed in general by A. H. Zimmerman of the division, and also by Dr. Hill, President Aeberly, Mr. Hart and Mr. Hattis. President Aeberly then called for a vote, the vote being unanimous in favor of the resolution.

John Howatt was then introduced as technical chairman of the discussion of Code Making. Mr. Howatt in his preliminary remarks gave the definition of a code and reasons for the developments of codes and how public demand, future growth and future development in the art must be anticipated, and cited various codes drawn up by the *A. S. M. E.* and the A. S. H. V. E.

The next speaker of the evening was Arnold H. Goetz, who stated that in other lines standard rather than specific requirements were set up. For instance in the pure food field, food must be kept in a certain condition but no specific method is set up as to how those conditions must be maintained.

S. R. Lewis, the first speaker for the specific requirements side, cited numerous state and other codes setting up specific requirements and the manner in which they had worked out.

Mr. Hart then read several codes in which specific requirements were set up but which were in effect nullified by requiring certain standards to be met.

J. A. Sutherland, explained various instances where on account of lack of information as to future occupancy it would be impossible to install a system to meet standard requirements.

Dr. Hill, the last speaker for the standard requirements, showed where the results to be obtained was really the vital question and the manner in which the engineer and contractor attained those results was immaterial.

Mr. Hale then read the list of members of the Code Committee of the A. S. H. V. E. and cited the fact that those eminent authorities had set up specific requirements in their code.

After considerable discussion on this subject took place, Mr. Haines presented a protest signed by 14 members present saying that they had not understood that the vote on the revised constitution was for final adoption and wished their votes recorded as opposed to its adoption. President Aeberly ruled that the protest was too late and that the protesting members would either have to make their protest to the National Council or submit the changes they wanted as amendments and follow the procedure set down in the constitution.

Ontario

December 7, 1931. Forty members and guests gathered in the Grill Room of the Prince George Hotel for the regular meeting of the Ontario Chapter.

W. R. Blackhall made a report on membership which was followed by an extended discussion. A special drive is being made to increase the membership of the Chapter.

M. W. Shears was appointed to take charge of transportation arrangements for the members going to the Annual Meeting of the Society in Cleveland, January 25-29.

The principal speaker of the evening was T. McDonald, who gave an illustrated talk on Automatic Temperature Control. Mr. McDonald was extended a vote of thanks by G. A. Playfair on behalf of the Chapter.

The meeting was then adjourned.

Philadelphia

December 10, 1931. Pres. E. N. Sanbern called the meeting to order in the Auditorium of the Engineers Club.

Following the reading of the minutes of the previous meeting, and the regular routine of business, on motion duly made and seconded, M. F. Blankin was elected as representative, and A. C. Edgar as alternate, to serve on the National Nominating Committee for 1932.

President Sanbern announced the appointment of the Auditing Committee as follows: A. A. Miller, A. C. Caldwell and J. H. Hucker.

F. D. Mensing spoke briefly, requesting as large an attendance as possible representative of the Philadelphia Chapter at the Annual Meeting to be held in Cleveland, January 25 to 29, 1932.

J. D. Cassell introduced the honored guest of the evening, W. H. Carrier, President of the Society, who in turn called upon A. V. Hutchinson, Secretary, who spoke briefly regarding the work that the Society has been carrying on, and outlined the events to take place at the Annual Meeting in Cleveland.

President Carrier presented a most interesting talk, illustrated with slides, on the subject of Servicing the Human Power Plant, in which he likened the human body to a modern power plant. He explained how temperature affected and acted on the human body, and showed the necessity for proper ventilation at certain temperatures to produce the desired relative conditions. He also outlined methods for determining the amount of air required for proper ventilation, and the necessity and importance of proper air distribution.

Several questions brought up by members were answered and discussed by President Carrier, after which the meeting was adjourned.

Pittsburgh

January 11, 1932. Pres. R. B. Stanger called the meeting of the Pittsburgh Chapter to order in McCann's Dining Rooms with 45 members and guests in attendance.

A short business meeting was held at which the minutes of the last meeting were read and approved.

E. C. Evans, chairman of the Membership Committee, spoke on membership in the Society and said that there were over 400 men in the Pittsburgh district who should be members of the A. S. H. V. E.

The chairman of the Program Committee, F. C. McIntosh, announced that he and his committee intended to work out interesting and worthwhile programs for the coming year.

On this occasion the usual formal address was dispensed with and the members and guests enjoyed a smoker and get-together. All were supplied with large corn cob pipes, a wealth of rough cut, genuine Pittsburgh stogies and accessory smoking equipment and spent the latter part of the evening talking over the "situation" with their friends. Several interesting reels of motion pictures were shown including a humorous reel on fishing and one on Carbon Monoxide, issued by the United States Bureau of Mines.

The party was adjourned at 10:00 p. m.

Southern California

January 11, 1932. A joint meeting of the Southern California Chapter of the A. S. H. V. E. and the *American Society of Refrigerating Engineers* was held at the Engineers' Club with an attendance of 97. The meeting was under the direction of L. H. Polderman, chairman of the Program Committee, and was devoted to the subject of Refrigeration.

The meeting was divided into two sections and the following men discussed fundamental principles of refrigerating equipment, illustrating their talks with slides and diagrams.

J. Laichinger and W. E. Barnum spoke on CO_2 Reciprocating Refrigerating Equipment.

M. M. Lawler and Chas. M. Barker discussed Methyl Chloride Rotary Machines.

Ammonia and Methyl Chloride Reciprocating Rotary Machines was the subject of the talk of C. E. Kilgore.

Louis R. Swenson spoke on Household Refrigeration.

Centrifugal Refrigeration was explained by L. H. Polderman and A. H. Simonds, Carrier Engineering Corp.

At the next meeting of the Chapter to be held February 15 the subject of Refrigeration will be continued. This meeting also promises to be very interesting and it is the hope of the Chapter that any Society members visiting in Southern California may be present to join in the discussions.

St. Louis

January 6, 1932. The St. Louis Chapter held a meeting, preceded by a dinner, at the Roosevelt Hotel at which 33 members and guests were present.

Pres. J. M. Foster called the meeting to order and the minutes of the previous meeting were read and approved.

The chairman of the Membership Committee, G. A. Helwig, read the applications of three new Limited Chapter Members and welcomed them into the Chapter.

L. Walter Moon, chairman of the Auditing Committee, stated that the books of the 1931 administration had been audited and approved and submitted the report.

President Foster appointed C. A. Pickett to represent the St. Louis Chapter on the National Nominating Committee, with F. J. McMorran as alternate, and instructed the secretary to certify their names to the Secretary's Office in New York.

President Foster also appointed E. B. Langenberg as counsellor and C. A. Pickett as alternate to assist J. M. Foster, chairman, and E. A. White, counsellor, for the Associated Engineering Society Committee and to represent the St. Louis Chapter of the A. S. H. V. E.

Walter Kaiser advised the Chapter that the Council of the City of St. Louis is now drafting a new building code and that an excellent opportunity exists to endorse a ventilating code with an attempt to have such a code adopted by the City. A motion was made by Mr. Moon to appoint a code committee to study the various City and State codes and report their findings at the next Chapter meeting. The motion was carried and President Foster appointed the following committee: L. Walter Moon, chairman, S. G. Hallett, and J. W. Cooper.

The meeting was then turned over to Paul Sodemann, chairman of the Program Committee, who introduced Earl H. Beling, research engineer, Herman Nelson Corp., Moline, Ill.

Mr. Beling addressed the Chapter on the subject of The New Science in Ventilation. He spoke briefly of the evolution of ventilation and then discussed the proper application of a modern ventilating system that would comply with the latest findings in the field of air conditioning research. The subject was very ably presented and heartily enjoyed by the members and guests present. Considerable discussion of the various theories presented followed and Mr. Beling answered many questions from those in the audience.

Archie Gordon, acoustical engineer, Hamilton Co., St. Louis, was then presented by Mr. Sodemann and gave a paper on Acoustics as Related to Heating and Ventilating. Mr. Gordon's subject was comparatively new to the membership and pertained to the isolation and elimination of several varieties of noises that occur in a ventilating system. Upon completion of the paper, the discussion indicated that the members had received both pleasure and benefit from the presentation of this subject.

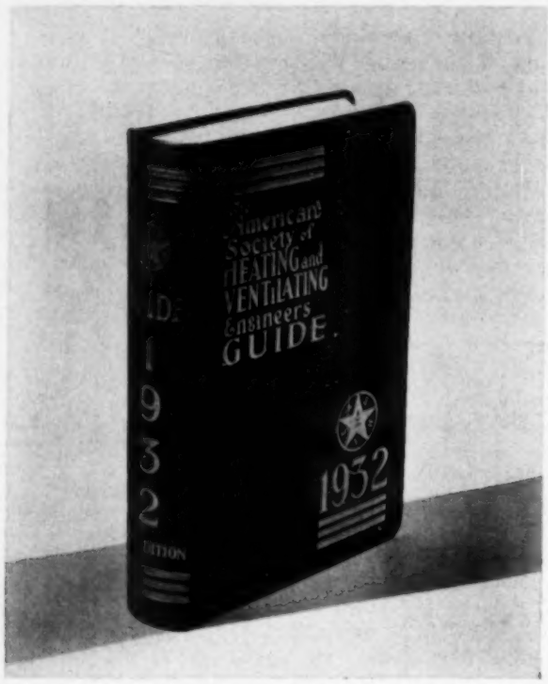
A vote of thanks was given to Messrs. Beling and Gordon for appearing before the meeting and presenting such interesting and instructive talks.

The A. S. H. V. E. Guide 1932

The 10th annual edition of THE GUIDE published by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 51 Madison Avenue, New York, N. Y., is just off the press. It appears in an entirely new binding—a light blue, flexible cover. Forty chapters of up-to-the-minute design and installation data on heating, ventilating and air conditioning, plus more than 300 pages of manufacturers' catalog data—amplified and revised to keep pace with important developments—have been recorded in this edition of THE GUIDE.

The outstanding features of THE GUIDE 1932 are as follows:

1. An increase in the number of chapters.
2. Approximately 50% new subject matter.
3. Every chapter revised or rewritten.
4. New grouping of subjects.
5. More extensive cross indexing to assist the user in locating needed specific information.



Chapter 1 contains an authoritative glossary and is intended to interpret the terms used in the Text Section. It is hoped that these definitions will serve to eliminate much of the confusion now existing concerning the lexicography of the heating and ventilating engineer. Comprehensive information on concealed heating units is given in a new chapter entitled Gravity Convectors. The Hot Water Heating chapter is entirely new and includes the latest results of researches on this subject. Information on devices for handling condensate and air is given in Chapter 10, whereas practical data on hot and cold water requirements of buildings, and the sizing of pipes therefor, will be found in Chapter 13.

The chapter on Pipe Insulation contains useful tables for quickly estimating the heat loss from pipes of various diameters covered with various thicknesses of insulation under a wide range of conditions. Entirely new information is included in the chapters on Draft and Chimneys, Mechanical Stokers, and Automatic Temperature Control. The subject of fuels has been expanded and valuable data on fuel requirements included. Many changes in the presentation of the subjects of ventilation and air conditioning have been necessary in order to incorporate the results of the latest research investigations. A distinction between ventilation and air conditioning has been made in the classification of the subject matter. Four chapters dealing with ventilation, and five dealing with air conditioning, have been included.

Chapter 36, Unit Ventilators, and Chapter 37, Unit Air Conditioners, are new and, with the exception of a small amount of previously published information, Chapters 38, 39 and 40 are new.

It is hoped that engineers, architects, contractors and others who use THE GUIDE 1932 will find in the 552 pages of technical data adequate information to solve the problems that arise in daily practice.

Modern equipment for practically any heating and ventilating service can be selected from the Catalog Data Section. Details of descriptions, sizes, capacities and dimensions are an invaluable supplement to the facts given in the Text Section. The manufacturers who make a careful and complete presentation give essential facts and figures on their specific products thereby amplifying the text. This unique method effectively covering both practice and equipment in a thorough and authoritative manner has gained for THE GUIDE worldwide recognition and acceptance as the standard reference work on heating and ventilating. —956 pages, flexible cloth—single copies $5.00.

Death of James H. Davis

On January 5, 1932, James H. Davis, one of the Society's Charter Members and also a Life Member, passed away in his 79th year. He had been a member of the Society since its organization in 1895, and served on many of its Committees and on the Council. He was one of the most active members in the establishment of the Society's Research Laboratory and continued his interest in this branch of the Society's activities throughout the active years of his life.

In his long and useful career Mr. Davis served in many branches of the heating and ventilating profession and was widely known and held in high esteem among his fellow engineers.

On October 12, 1853, James H. Davis was born at Oswego, N. Y., and after receiving his schooling entered the steam heating business as an apprentice with Baker Smith & Co., Chicago. He was with this company from 1878 to 1890, serving successfully as estimator, designer, superintendent and secretary, and was familiar with all the operations of designing and erecting steam and hot-water heating and ventilating plants.

Later he served with several heating contracting firms, F. W. Lamb & Co., Chicago, Missouri Steam Heating Co., St. Louis, and the American Heating Co., St. Louis.

During his many years of service with both Warren-Webster Co. and American Radiator Co. he witnessed the growth of Chicago, and supervised the design and installation of heating and ventilating plants in many famous old buildings such as the original Palmer House, the Pullman Building, Chamber of Commerce and a score of others.

Most of his business life was spent in Chicago and since his retirement from active work he has remained in the city.

He is survived by his widow, Emily E. Davis, and a son, Richard J., and a daughter, E. Olive Davis, to whom the Officers and Council of the Society extend their sincere sympathy for the loss they have sustained.

At Mr. Davis' funeral the Society was represented by its Past-President, John F. Hale, Chicago, and at the meeting of the Local Chapter, of which Mr. Davis was a Charter Member, appropriate resolutions on his death were adopted.

Heating and Ventilating Course at N. Y. U.

A certificate evening course in heating and ventilation is given at New York University. This course is designed to supplement the practical experience of those now employed in the design, erection and operation of heating and ventilating installations, as well as to prepare those who are about to enter this field. It includes heating systems; chemistry of air, water, metals, and fuels with laboratory practice; theory of heat with laboratory work; principles of heating and ventilating design; applications

of heating and ventilating design; heating and ventilating testing; estimating principles and commercial practices; and mechanical and electrical equipment of buildings.

The curriculum is under the direction of an advisory committee consisting of A. A. Adler, W. H. Carrier, W. H. Driscoll, W. L. Durand, W. W. Hubbard, H. C. Meyer, Jr., T. S. Tenney and C. M. Wooley.

The first session of the first term of each certificate class is held on a scheduled evening during the second week after Labor Day week. The first session of the second term is held on its scheduled evening during the first week of February. Preparation equivalent to high-school graduation is required for certificate courses. Students with less preparation may be admitted provided they have sufficient maturity of experience.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 8 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by February 15, 1932, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BLACK, WILLIAM BOGARDUS, Salesman and Engineer, Bryant Gas Heating, Inc., Pittsburgh, Pa.	C. W. Wheeler H. E. Digby	H. G. Bucher D. J. Rank
FENTON, FREDERICK A., Asst. Br. Mgr., Richardson & Boynton Co., New York, N. Y.	H. W. Hochuli L. G. Paulding	H. R. Dillon R. C. Willis
KEELER, ANSON HOWARD, Sales Engr., Vento & Arcoblast Dept., American Radiator Co., Philadelphia, Pa.	H. G. Kreissl H. R. Linn	C. H. Quirk Harvey Rettew
PAUL, DONALD IRWIN, Gurney Foundry Co., Toronto, Ont., Canada.	W. R. Blackhall G. T. Wilson	A. W. Wilson William Philip
REID, HERBERT F., Partner, Reid-Graff Co., Muskegon Heights, Mich.	W. W. Bradfield O. D. Marshall	D. L. Taze P. J. Busch
ROCKWELL, THEODORE F., Instr., Carnegie Institute of Technology, Pittsburgh, Pa.	F. B. Rowley A. B. Algren	J. L. Blackshaw F. C. Houghten
WALDON, CHARLES DENCHFIELD, Sales Mgr., Spencer Foundry Co., Ltd., Toronto, Ont., Canada.	W. R. Blackhall S. J. Brooks	William Philip G. E. Cole
WALSH, CLIFFORD J., Johnston Bros., Inc., Ferrysburg, Mich.	J. F. Johnston, Jr. N. F. Yonkman—*Non-Member*	P. J. Busch D. L. Taze

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

DOWNS, SEWELL HENRY, Chief Engineer, Clarage Fan Co., Kalamazoo, Mich.

HULL, HARRY BLAIR, Frigidaire Corp., Dayton, Ohio.

REID, HENRY P., Universal Atlas Cement Co., Chicago, Ill. (*Advancement*)

VOGELBACH, OSCAR, Chief Engineer, Guilbert & Betelle, Newark, N. J. (*Reinstatement*)

ASSOCIATES

ARNOLD, EDWARD Y., Secretary-Manager, Heating and Piping Contractors St. Paul Assn., St. Paul, Minn.

CHRISTENSON, HARRY, Heating Superintendent, Hunter Prell Co., Battle Creek, Mich.

MCDONALD, THOMAS, Canadian Manager, Minneapolis-Honeywell Regulator Co., Toronto, Ont., Canada

NIGHTINGALE, GEORGE F., Contract Sales Mgr., Hart & Cooley Mfg. Co., Chicago, Ill.

Heating · Piping and Air Conditioning

EDITORIALS

» » There is little doubt that the piping of an industrial or power plant too often receives the least consideration of the major items involved. Piping forms the connecting links in the chain of apparatus that turns out the product—whatever it may be. If the links are not properly installed or not properly taken care of after installation, the chances for serious troubles are ever-present.

The selection and installation of the producing apparatus invariably receives a great deal of careful consideration. However, this equipment cannot operate economically and efficiently if the piping connections are not properly installed. It is never good practice to purchase cheap piping material or cheap piping service. The designing engineer does well to remember that the competence of the piping contractor should be a major consideration when installing a piping job. The operating engineer is concerned with the piping being installed so that maintenance will be a minimum. The cost of steam leaks, radiation losses due to broken covering, improper flow due to poor design, etc., run into appalling figures, and careful maintenance, which is facilitated by proper installation, should never be disregarded.

We are now in a period which calls for the highest efficiency in operation. Some plants are inclined to make the serious mistake of neglecting maintenance of the piping systems. This is surely false economy. It would seem that the proper thing to do would be to keep the plant, particularly the piping, in first-class condition.

» » Newspapers and general magazines are devoting an increasing amount of space every day, it seems, to the subject of air conditioning for human comfort and health. Articles addressed to the general public stress the benefits of comfortable and healthful air; the financial pages are concerned mainly with the effect the expected development of air conditioning equipment and systems will have on our business activity; many of these articles are of particular interest to management and to landlords, as they point out what air conditioning means to efficiency and to securing and keeping tenants.

The air conditioning engineer, trained as he must be in the technical aspects of the air conditioning problem, may be inclined to look upon some of these articles with disfavor, as frequently an incomplete—or even incorrect—comprehension of the subject is indicated. It would seem, however, that slight technical inaccuracies in air conditioning articles in general publications, which are of course addressed to the layman, should not impair the value of this publicity.

When the general public realizes fully what air conditioning can do in providing comfort and health, demand for it is bound to increase. A survey of general publications and newspapers would indicate that this fuller realization is being rapidly developed.

» » "I am often asked the question, 'What should be and what should not be standardized?' " says P. G. Agnew of the *American Standards Association*. "My standard answer is, that standardization should be applied only to those fields in which it can fill a real human need. In industry and in technology standardization should be applied only to those things in which its application will result in an economy of human effort or in an economy of materials. We should not standardize anything merely for the sake of uniformity—that is deadening."

Seen from this view-point, standardization becomes a dynamic industrial tool. Consider, for example, its application to heating, piping and air conditioning. The industry as we know it today has been made possible to a large extent by standardization. Just as words and letters are the bricks out of which languages and literatures are constructed, so the industry in its present form owes its existence to such standards and units as pipe, pipe threads, valves, tees, and unions, and to scores of other standard parts and materials that have been created in the course of its development.

Sound standardization wisely carried out in any of the numerous fields to which it is applicable is a means of furthering and of serving the results of pioneering research and development in technology, industry, and business. It is more. It is, in a sense, a liberator, since it reduces industrial advances to the field of routine, thereby clearing the way for further progress in research and development. « «

Practical Piping Problems

Unloading Oil From Tank Cars

W. H. Wilson's Monthly Discussion of the Problems He Meets From Day-to-Day

Unloading Oil Car with Defective Valve: Occasionally oil tank cars are received with defective valves on the unloading nozzles and the contents of the car cannot be unloaded in the usual manner. It is not always convenient to pump out the contents through the opening in the dome on the top of the car. The writer used the following method a few days ago.

Fig. 1 shows the location of the unloading nozzle *A* which is equipped with a cap *C* that is normally screwed on tight. Such a cap is generally provided with a 2-in. pipe thread tapping in the bottom, and a plug.

Procure a 2-in. angle valve *D* and remove its stem and disc. Take a rod *E* of approximately the same diameter as the stem and insert it in the valve by passing it through the packing box. Remove the 2-in. plug and push the rod up in the unloading nozzle. Attach the angle valve to the opening in the bottom of the cap by a close pipe nipple. The length of the rod should be the length of the unloading nozzle, plus the distance the cap projects below the nozzle, the length of the exposed part

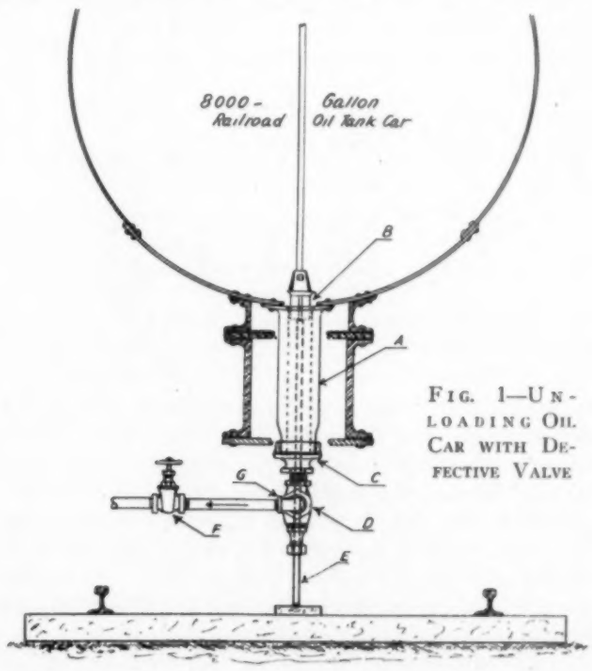


FIG. 1—UNLOADING OIL CAR WITH DEFECTIVE VALVE

of the pipe nipple and the length of the angle valve from the inlet end to the end of the packing box. In the case illustrated, the length of the rod was 40 inches.

It is usually necessary to make a hole in the ground between the ties of the track in order to get enough room to insert this rod in its proper position. By using a close pipe nipple and an angle valve of a short pattern the rod can be put in place conveniently. In case it is not convenient to get this room below the ground level, the rod can be made in two pieces.

Install a 2-in. pipe from the side opening of the angle valve to the oil storage tank. By using an elbow, as shown at *G*, the pipe line can be inclined downward or more easily adjusted for the required unloading position.

When ready to unload, tighten the packing box of the valve around the rod, open the dome cover on the top of the car and force the rod upward until the oil valve *B* is raised from its seat the distance required for full flow and block up the lower end of this rod so it will remain in this position.

A gate valve is placed in the oil line at *F* in case it is necessary to stop the flow of oil. When unloading is completed the angle valve can be removed, the stem and disc replaced and the valve used for any of its usual requirements.

Steam Heater for Oil Car Unloading Nozzle: Heat is required for the unloading of tank cars containing heavy oils. In case these cars are not equipped with heater pipes, a two-section pipe coil can be inserted through the dome opening on top of the car, or the following arrangement can be inserted in the unloading nozzle.

Fig. 2 shows the location of the unloading nozzle *J*, the unloading hose *M*, and the elbow *N*.

By welding in a nipple with a union connection and packing box, as shown at *P*, the tube *K* can be inserted through this opening and extended in the unloading nozzle. This tube, which can be a 1½-in. pipe, is closed on the upper end by welding. The lower end is equipped with a tee *R*. A ¾-in. pipe *L* is inserted through the tee

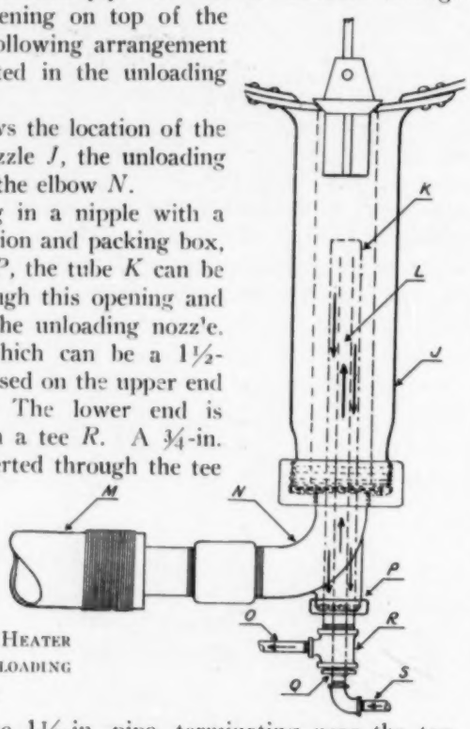


Fig. 2—Steam Heater for Oil Car Unloading Nozzle

up through the 1½-in. pipe, terminating near the top with an open end. Where the ¾-in. pipe passes through the bushing at the lower end of the tee (*Q*) this joint may be made by welding.

Steam is admitted at *S* and passes up through the ¾-in. pipe; it flows back and downward with the condensation through the 1½-in. pipe, passing out the side opening of the tee *O* where a suitable steam trap may be installed.

The effective heating surface of this arrangement depends on the length of the heating pipe in the unloading nozzle, which should extend into the nozzle as far as possible.

Recent Trade Literature

Conveying: The Brady Conveyors Corporation, 20 W. Jackson Blvd., Chicago, Ill.; 12-page booklet giving general descriptions of pneumatic pressure and suction systems, car and boat unloaders and loaders, portable pneumatic systems, ash handling systems, lime unloaders for water plants, marine legs and dust collectors.

Heating Specialties: Sarco Company, Inc., 183 Madison ave., New York City; 16-page catalog illustrating and describing typical vacuum and vapor heating systems and describing thermostatic specialties such as radiator traps, packless valves, receivers, air eliminators, etc.

Heating Systems: Warren Webster & Company, Camden, N. J.; 20-page booklet describing the operation of vacuum heating systems with metering orifices and without orifices and gravity two-pipe steam systems with and without orifices. Control, zoning, flexibility, etc., are discussed and the necessary equipment is described in detail, with typical specifications.

Materials Handling: Link-Belt Company, 910 S. Michigan ave., Chicago, Ill.; calendar illustrating coal and ash, sand and gravel, etc., handling systems and drives for power transmission.

Motors: Westinghouse Electric & Manufacturing Company, Nuttall Works, Pittsburgh, Pa.; four-page data sheet describing speed reducers combined with induction motors (½ to 15 hp., 69 to 1550 r.p.m.) and discussing applications, construction, output speeds, etc., and giving dimensions and a table of engineering information.

Separators: The Leavitt Machine Co., Orange, Mass.; six-page folder describing automatic separators for removing oil and water from compressed air lines; one type where pressure regulation is not necessary and one used when pressure regulation is needed, as in spray painting. Also a folder describing construction and applications of receiver drains; price lists are given.

Stokers: Holcomb & Hoke Mfg. Co., Indianapolis, Ind.; reprint describing the function of the hopper gas eliminator, which also maintains a balanced draft condition.

Unit Heaters: The Herman Nelson Corporation, Moline, Illinois; 48-page catalog describing and illustrating the features of construction of suspended type unit heaters for industrial and commercial buildings, warehouses, etc. Dimensions, capacity tables, recirculating ducts, high pressure steam, control, piping suggestions, wiring information, specifications, etc., are among the topics presented.

Vacuum Cleaners: Arco Vacuum Corporation, Division of American Radiator Company, 40 W. 40th st., New York City; four-page bulletin giving specifications, price and description of portable industrial type vacuum cleaners equipped with 1½-hp. motors.

Welding Equipment: Air Reduction Sales Company, Lincoln Building, New York City; 26-page pocket-size catalog listing specifications and illustrating welding and cutting torches, regulators, cylinder trucks, hose connections, etc. Mechanically-operated machines for cutting are also described.

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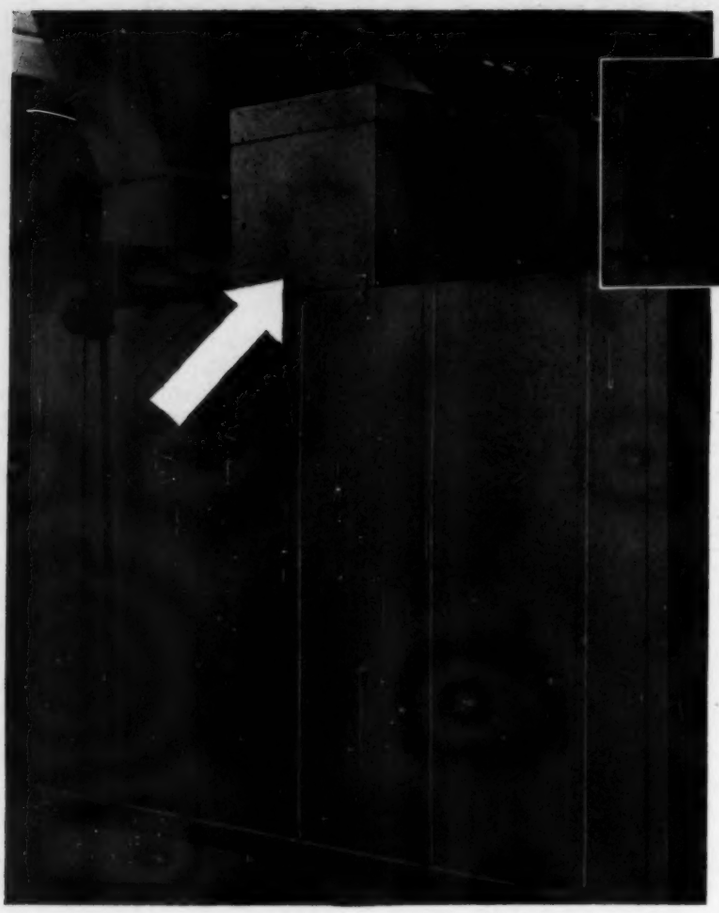
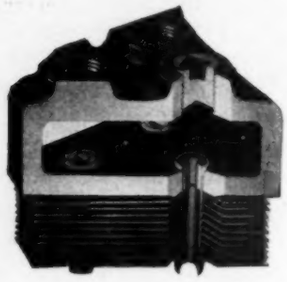


A typical Trane Extended Surface Heating Element

Industrial ovens heated with Trane Extended Surface Elements. Arrow shows position of heating units.



Sectional view through header of Trane Extended Surface Unit. Note the cut open tube showing construction — integral fit of extended fins to tube and orifice distribution bushing.

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TRANE

UNIT VENTILATORS

The EDITOR'S PAGE

Heating · Piping and Air Conditioning

The air conditioning system in the New York Curb Exchange building is arranged in such manner as to enable economies by careful operation. For instance, the trading floor system is in use until 3 p. m., while the clearing house system may be required to be in operation until

late in the evening; a cold water storage tank is therefore provided to furnish cold water for the clearing house equipment after 3 p. m. This eliminates the necessity of operating the refrigeration machine at a light load and consequent low efficiency.

Chester L. Nelson has described the conditioning equipment for this building in our lead-off article; the trading floor in the Exchange is one of the largest in the country.

Many of the speakers at the air conditioning conference to be held at *Case School of Applied Science* this month are well-known to the readers of HEATING, PIPING AND AIR CONDITIONING. A number of interesting-looking papers are to be presented, and the sessions are planned to cover principles of air conditioning, the unit processes and their control, health and comfort, and applications and economics.

The conferences, which are sponsored by Case and the *Cleveland Engineering Society,* are designed to afford the engineer a means of continuing his education. W. E. Wickenden, president of Case, summed up the need of the engineer for continually keeping in touch with new processes and developments in his opening remarks at a previous meeting. These are quoted in the editorial in this issue.

A mile-long underground line for conveying steam for process and power has been constructed by the Indianapolis Power & Light Co. to supply the Eli Lilly Company; a ten-year contract recently signed provides that steam (65,000 lb. per hr.) is to be delivered at 175 lb. and 100 F superheat. The steam will be used to generate electricity at the plant which will be fed into the utility's lines, the exhaust steam being used for process work. The construction of this line—the first of record required to deliver steam at 100 F superheat a mile from the boilers—is described by Robert L. Fitzgerald in this issue.

Direct-fired gas heaters are finding wider application in many localities; in this issue Platte Overton shows how the direct-fired heater for the offices of an industrial plant was selected and how the system was designed. Summer cooling is provided by passing city water through two sections of coils in the duct; in winter, the air is humidified.

E. H. deConingh wrote a letter to us last month from which we hasten to quote. "I do not know of any other technical journal among the many we receive," says Mr. deConingh, "which covers a field in as thorough and readable manner as does HEATING, PIPING AND AIR CONDITIONING."

WATCH FOR THESE NEXT MONTH

John J. Harman writes on testing methods for pipe joints in high pressure steam lines in the April number. A logarithmic chart showing turning moments required to produce various unit stresses in bolts of different sizes is of particular interest.

The third installment of Sabin Crocker's "Selecting the Right Size Heating Boiler" will be published in April. It discusses grate area and fuel burning rate, combustion space in furnace, type of heat liberation and the effect of these on sizing the boiler.

Malcolm Tomlinson has prepared a paper on calculations for drying which you'll find in next month's issue. Practical examples and charts make it of great value.

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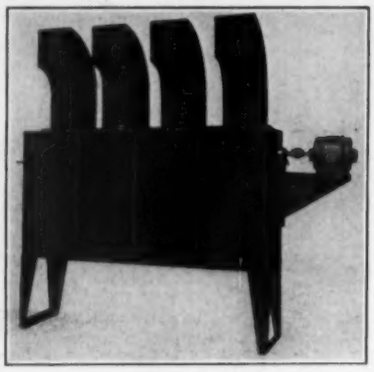

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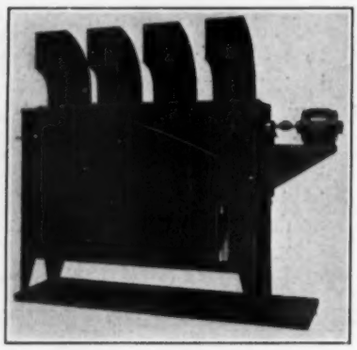

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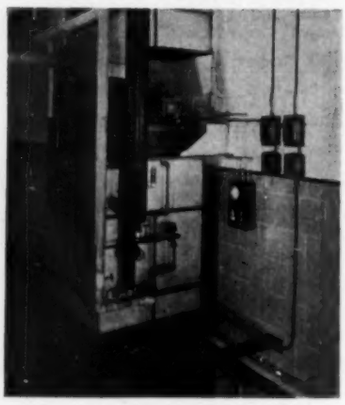

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Published Monthly by **Engineering Publications, Inc.** 1900 Prairie Avenue, Chicago

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Current copies, 50c each. Back numbers, $1.00 each.

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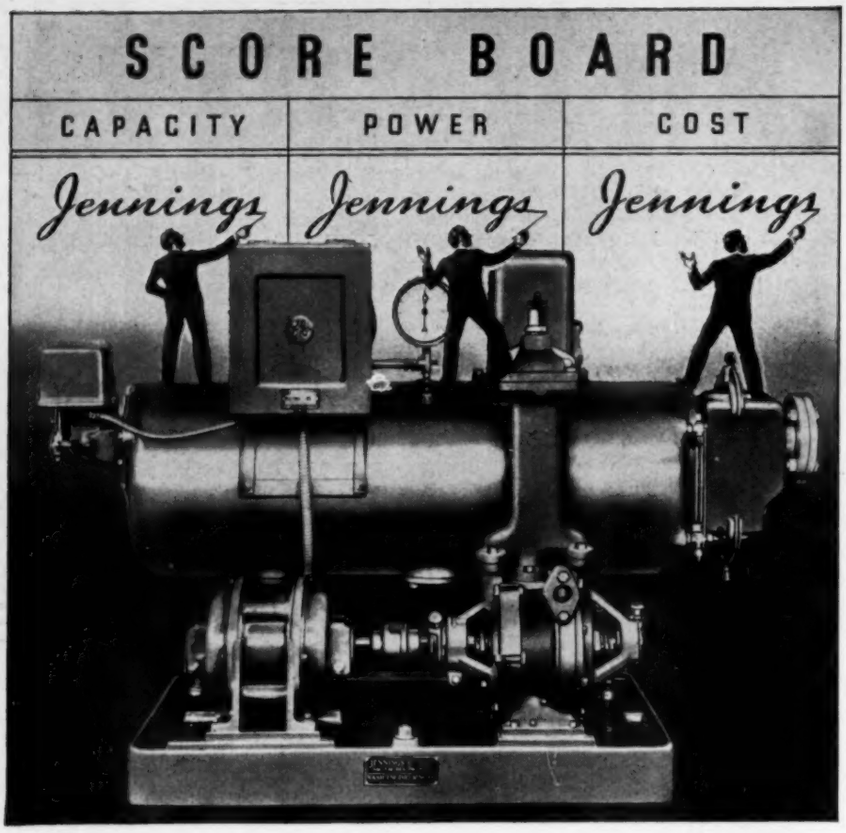

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